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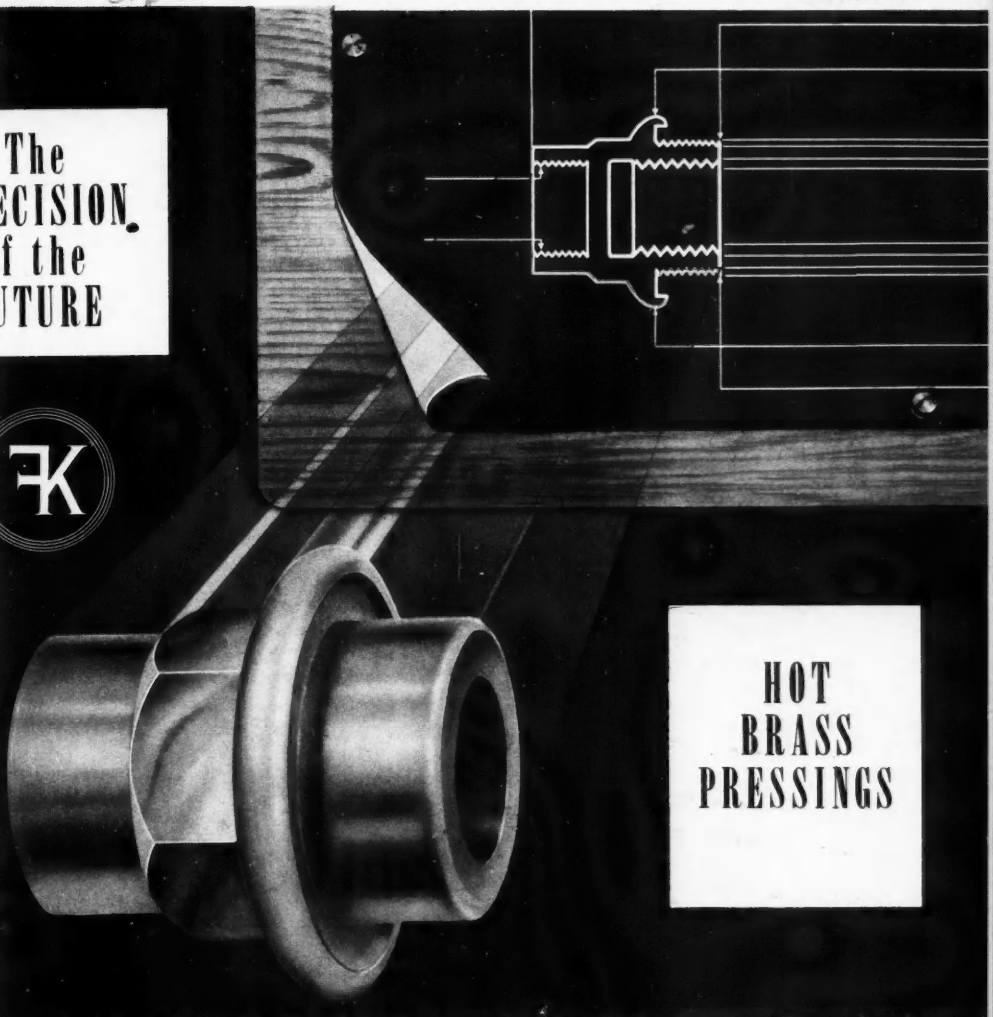

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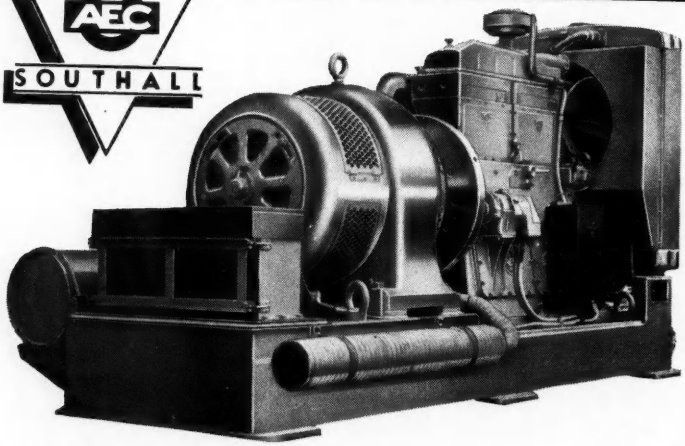

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THE ENGINEERS' DIGEST

REVIEW OF WORLD ENGINEERING PROGRESS

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IN THIS ISSUE

THE article by E. Niedermayer on rheological criteria for two-stroke cycle scavenge systems is a continuation of the research work undertaken by List, Hold, Pischinger, Reyl, Schultz-Grunov, and other investigators, on the assessment of the scavenge performance of engines and blowers. The first part of this study, reproduced in this issue on the following pages, reviews the fundamental criteria for two-stroke cycle machines, and relates the determination of optimum conditions for a given system to the study of its variations in performance with various slot dimensions and arrangements, and air supply and discharge coefficients, on the basis of observations made on a model. Formulae are given which enable the results of the model tests to be utilised for the prediction of the characteristic parameters and overall efficiency of the design considered. In this work, two-stroke cycle engines, Roots blowers, and centrifugal blowers are investigated separately, and also studied when so combined as to form a pressure-charged scavenge system.

* * *

The article by Liselius on balancing, see page 161, shows how a complete static and dynamic balancing of rotors can be achieved by introducing additional forces in two arbitrarily selected balancing planes. It also describes two types of balancing machines, the first with an oscillation point, and the second with freely oscillating bearings. Among the interesting features of these machines there might be mentioned a new vibration pick-up of the capacitive type, and the phase-adjusting arrangement used to determine the heavy section of a rotor.

* * *

The question of standardisation of steam turbines is not easily solved, as turbines are inherently machines which have to be designed to fit in with a certain set of conditions and, naturally, each individual set of conditions will require a special design if high efficiency is to be obtained. The question of efficiency, of course, assumes even greater importance because of the rising costs of fuel all over the world. On the other hand, shortage of generating plant demands higher production and, with existing limitations of production facilities, this can only be achieved by standardisation. Some time ago a German plan for standard power stations was described in this journal*; this plan, however, was interrupted before it could be carried out. In this issue, the abstract on page 157 dealing with Russian views on standardisation may serve as a comparison with British and American practice. It will be seen that, whereas in this country only two ratings have been adopted for public electricity supply, leaving industrial turbines entirely free, in the U.S.S.R. 57 types and ratings have been included in their standard specifications, covering straight condensing, extraction and back-pressure turbines. It must be remembered, however, that the proportion of extraction turbines in Russia is much higher, owing to the widespread use of district heating, and that a large number of back-pressure turbines will be superimposed on existing installations with a view to improving efficiency.

*THE ENGINEERS' DIGEST, Vol. IX, No. 1, January, 1948, p. 13.

* * *

Steam turbines, in general, are classified in two groups, viz., condensing and non-condensing or back-pressure turbines, terms which refer to the turbine exhaust pressure. A special type comprised by the latter group is the superimposed or topping turbine which has been widely used in recent years for the extension of existing power plant, especially in Germany, Russia, and the U.S.A. Turbines of this type are designed to operate under higher initial steam conditions and to exhaust against the inlet pressure of the existing turbines. Thus, a thermal efficiency is achieved better than that which would have been obtained if another condensing unit had been added.

In an article taken from the house journal of the Swiss firm Oerlikon, see page 158, an interesting proposal is put forward for increasing the generating capacity of industrial power plant, consisting of a back-pressure turbine, the exhaust of which is used for process steam or space heating. The authors advocate "superimposing" a gas turbine on the existing thermal cycle, and show, by way of numerical examples, that considerable fuel saving can be effected in comparison with the somewhat more orthodox procedure of extending the plant by a condensing turbine. We feel justified in using the term "superimposed" here for want of a better expression, and also because a certain similarity exists between this compound cycle and the system of "topping" outlined above.

Rheological Criteria for Two-Stroke Cycle Scavenge Systems

By E. NIEDERMAYER. (From *Maschinenbau und Wärmewirtschaft*, Vol. 4, Nos. 1 and 2, January and February, 1949, pp. 1-7 and 18-24, 11 illustrations.)

A. SCAVENGE PARAMETERS

THE gas-exchange process, which comprises the scavenging and induction processes occurring between the opening of the exhaust slots and the closing of the inlet, can be fully characterised by three parameters, viz. :—(1) the air supply ratio λ_l; (2) the scavenge ratio λ_s, and (3) the air utilization factor S.

The *air supply ratio* is defined as the volume of the fresh charge L_z available in the cylinder after renewal of the air charge, divided by the stroke volume (i.e., the swept volume) V_h:

$$\lambda_l = L_z/V_h = \frac{\text{fresh charge}}{\text{stroke volume}}.$$

The *scavenge ratio* is a measure of the quality of the scavenging. It is defined as the ratio of the fresh charge at the end of the gas-exchange process to the total charge in the cylinder at this same instant:

$$\lambda_s = L_z/(L_z + R_z) = \frac{\text{fresh charge}}{\text{total charge}}$$

R_z denoting the residual gases in the cylinder after scavenging, the residual gases having the same composition as the combustion products in the cylinder at the beginning of the scavenging process.

The *air utilization factor* indicates the total quantity of scavenge air (i.e., the total quantity of air, or fuel/air mixture, L_g, delivered through the inlet orifice) required to obtain a given scavenge ratio λ_s. It is defined as the ratio of the total scavenge air L_g to the stroke volume:

$$S = L_g/V_h = \frac{\text{fresh charge} + \text{scavenge air}}{\text{stroke volume}}$$

All volumes are expressed in cubic metres per working cycle, and referred to prevailing external atmospheric conditions.

Furthermore, in mixture-scavenged engines the fuel loss can be determined by means of the ratio

$$\lambda_z = L_z/L_g = \lambda_l/S = \frac{\text{fresh charge}}{\text{fresh charge} + \text{scavenge air}}$$

which is known as the *charge factor*.

In addition, it is frequently useful to express the total charge by means of a ratio which relates the total charge in the cylinder at the end of the gas-exchange process to the stroke volume:

$$\lambda_g = (L_z + R_z)/V_h = \lambda_l/\lambda_s = \frac{\text{total charge}}{\text{stroke volume}}$$

The scavenging is affected by two processes which are not independent of one another, viz.: (1) the more or less complete mixing of the scavenge air with the residual gases, and (2) the heat exchanged between these two volumes. In general, the interchange of heat due to thermal conduction between adjacent gas layers is insignificant compared with the heat transfer due to diffusion of the fresh air with the combustion products. Therefore, a definite value for the mean charge temperature can be related to a given quantity of residual gases in the cylinder, irrespective of the particular scavenge process considered.

Assuming a constant charge pressure during the scavenging process, a volume of gas defined by reference to prevailing external conditions will, on entering the cylinder, displace an equal volume of combustion products determined with reference to the charge temperature. Furthermore, owing to the difference in the specific heats of the air and the residual gases, the interchange of heat will cause an increase in the volume of the charge, so that with a constant charge pressure the volume of gas of the efflux will be greater than that of the air intake. The resulting increase in scavenge efficiency can reach 10 per cent in some cases. However, it will not be taken into account in the following considerations of the basic principles of two-stroke cycle systems.

1. AIR SUPPLY RATIO.

With the assumption mentioned above, the air charge in the cylinder at the end of the scavenging period is independent of the residual-gas temperature before scavenging, regardless of the type of scavenging considered.

Thus, for positive-displacement scavenging

$$L_z = L_g = S\,V_h$$

and the air supply ratio is

$$\lambda_l = S \qquad .. \qquad .. \qquad .. \qquad (1)$$

In the case of negative-impulse scavenging, the alteration of the air charge in the cylinder is given by

$$dL_z = dL_g - dL_g\,\frac{p_z\,T_e}{p_z\,T_z}\,\frac{L_z\,p_a/T_a}{V_z\,p_z/T_z}.$$

In this equation, V_z = mean cylinder volume during scavenging, p = gas pressure, and T = gas temperature (°C abs.). The subscript a refers to external conditions, the subscript e to the condition of the scavenge air just before entering the cylinder, and the subscript z to conditions in the cylinder. The integration gives

$$L_z = \left(1 - e^{-\frac{V_h\,p_a\,T_e}{V_z\,p_z\,T_a}}\right) V_z\,\frac{p_z\,T_a}{p_a\,T_e},$$

and the air supply ratio is, therefore,

$$\lambda_l = \frac{V_z\,p_z\,T_a}{V_h\,p_a\,T_e}\left(1 - e^{-S_z}\right). \qquad .. \qquad .. \qquad (2)$$

2. SCAVENGE RATIO.

To assess the effects of various arrangements of the scavenging slots and various flow velocities on the quality of the scavenging process, it is advisable to use a model having the same dimensions as the engine cylinder to be scavenged. With this model it will be possible to follow the scavenging process, and to measure the scavenge ratio.

The gas temperature in the model, before scavenging, is equal to that of the scavenge air. The residual-gas volume in the engine, at the commencement of scavening, referred to external conditions, is thus smaller than the gas volume in the scavenge model at the same instant. Thus, under otherwise similar conditions, the scavenge ratio of the engine will be greater than that of the model.

For positive scavenging, the scavenge ratio of the model is

$$\lambda_{sM} = L_z \Big/ \left(V_z\,\frac{p_z\,T_a}{p_a\,T_e}\right),$$

where the subscript M refers to the model. Substitution of the air utilisation factor in this equation gives

$$\lambda_{sM} = S\,\frac{V_h\,p_a\,T_e}{V_z\,p_z\,T_a} = S_z. \qquad .. \qquad .. \qquad (3)$$

In the case of negative-pulse scavenging, the scavenge ratio for the model is

$$\lambda_{sM} = \frac{L_z}{V_z \dfrac{p_z\, T_a}{p_a\, T_e}} = 1 - e^{-S_z} \qquad .. \quad (4)$$

For both types of scavenging, the relationship between the scavenge ratio of the model and that of the engine is given by

$$\lambda_s = \frac{L_z}{L_z + \left(V_z - L_z \dfrac{p_a\, T_e}{p_z\, T_a}\right) \dfrac{p_z\, T_a}{p_a\, T_{ze}}}$$

or on substituting λ_{sM}:

$$\lambda_s = \lambda_{sM} \Big/ \left[\lambda_{sM}\left(1 - \frac{T_e}{T_{ze}}\right) + \frac{T_e}{T_{ze}}\right], \quad .. \quad (5)$$

T_{ze} being the residual-gas temperature at the beginning of scavenging. T_{ze} can be calculated from the charge factor and the expansion ratio during the preliminary exhaust period, neglecting the change in weight of the charge due to the fuel injected:

$$\lambda_v V_h p_a/T_a = V_{za} p_{za}/T_{za} = V_{za} p_{za} P^n/T_{ze}$$
$$T_{ze} = T_a V_{za} p_{za} P^n/(V_h \lambda_v p_a) \quad .. \quad .. \quad (6)$$

where $P = p_{ze}/p_{za}$ and $n = (m-1)/m$, m being the polytropic exponent for the change of state during the preliminary exhaust period, while za and ze denote the state of the charge before the preliminary exhaust and at the beginning of the scavenging period, respectively.

The scavenge ratios λ_s and λ_{sM} are plotted as functions of the air utilisation factor S_z in Fig. 1, for various

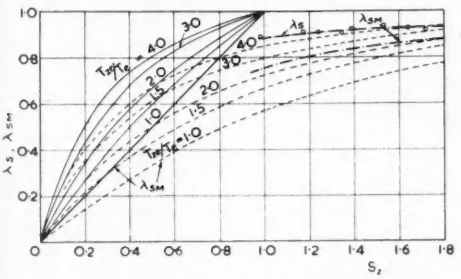

Fig. 1. **Scavenge ratio curves of the model and engine for various values of the ratio T_{ze}/T_e, (temperature of residual gases divided by air temperature).**
Solid lines = Positive scavenging.
Dotted lines = Reverse scavenging.
(Circles indicate measured values of the engine).

values of the parameter T_{ze}/T_e, calculated on the basis of eqs. (3), (4) and (5). Measured values of λ_{sM} for a model with reverse-flow scavenging, and measurement points relating to an actual engine, are also indicated by chain-dotted lines in this same figure. The curves show that eq. (5) gives with sufficient accuracy the relationship between the two scavenge ratios.

It is also apparent from Fig. 1 that high scavenge values are attainable even with negative scavenging at high residual-gas temperatures, with a slightly reduced total charge. The total charge can be expressed as a function of the scavenge ratio of the model and of the ratio of air-charge temperature to residual-gas temperature:

$$\lambda_v = \frac{V_z p_z\, T_a}{V_h p_a\, T_e}\left[\lambda_{sM}\left(1 - \frac{T_e}{T_{ze}}\right) + \frac{T_e}{T_{ze}}\right]. \quad .. \quad (7)$$

3. Air Utilisation of Scavenge Air Blower.

The mean effective pressure p_e of an engine can be calculated from the air supply ratio λ_l, the scavenge ratio λ_s, the fuel consumption per horsepower-hour b_e, and the excess air ratio λ_v of the combustion system under consideration. Following H. List,* we have

$$p_e = \frac{27}{L_0\, b_e} \cdot \frac{\lambda_l}{1 + (\lambda_v - 1)\,\lambda_s} - p_g,$$

where L_0 = theoretical volume of air (m³) required per kg of fuel, and p_g = mean pressure (kg/cm²) assumed to be acting on the piston surface, which would be required to drive the scavenge blower during one stroke.

Using eq. (5) for λ_s and eqs. (2) and (4) for λ_l, and substituting

$$\lambda_l = \lambda_{sM}\, V_z p_z\, T_a/(V_h p_a\, T_e) = \lambda_{sM}\, t_1$$

we obtain

$$p_e = \frac{27}{L_0\, b_e} \cdot \frac{\lambda_{sM}\, t_1}{1 + (\lambda_v - 1)\dfrac{\lambda_{sM}}{\lambda_{sM}(1 - t_2) + t_2}} - p_g \qquad (8)$$

where $t_1 = V_z p_z\, T_a/(V_h p_a\, T_e)$ and $t_2 = T_e/T_{ze}$. The excess air ratio λ_v also takes account of the air contained in the residual gases.

If $p_e + p_g$ is plotted against λ_{sM} according to eq. (8), for various values of T_{ze}/T_e, it is seen that the mean pressure is only slightly affected by the ratio of residual gas temperature to air-charge temperature, and is, therefore, primarily proportional to the scavenge ratio λ_{sM} of the model. This shows, therefore, that the scavenge ratio is the primary parameter to be considered in assessing the effectiveness of a scavenge system.

The power absorbed by the scavenge blower, expressed as a mean pressure, is given by

$$p_g = S \gamma_a H_{ad}\, 10^{-4}/\eta_{ad} \quad [\text{kg/cm}^2] \quad .. \quad (9)$$

where γ_a = density of air entering the blower [kg/m³], η_{ad} = overall adiabatic efficiency of the blower, and H_{ad} = total adiabatic delivery of the blower, determined by means of the relation:

$$H_{ad} = \frac{p_a}{r\,\gamma_a}\left[(p_s/p_a)^r - 1\right] + (C_a^2 - C_0^2)/(2g) \quad .. \quad (10)$$

In this equation, $r = (k-1)/k$, k denoting the ratio of specific heats, p_s = delivery pressure, and C_a, C_0 = air velocities in pressure and suction ducts of the blower, which in most cases can be regarded as being equal.

The overall adiabatic efficiency η_{ad} varies, according to the qualities of the design and the position of the operating point in the blower diagram, within the following limits:

Type	adiab. efficiency
Piston-type blower	$\eta_{ad} = 0{\cdot}95 - 1{\cdot}00 \times \eta_{mech}$
Roots blower ..	$\eta_{ad} = 0{\cdot}40 - 0{\cdot}70 \times \eta_{mech}$
Centrifugal blower	$\eta_{ad} = 0{\cdot}50 - 0{\cdot}76 \times \eta_{mech}$

Fig. 2 gives the characteristic curves of a Roots blower with a rotor diameter of 180 mm and a rotor clearance of 0·2 mm, as well as the curves for a single-stage radial blower with a 150 mm rotor and a spiral outlet. The dotted lines marked *a*, *b*, and *c*, also included in Fig. 2, represent three different types of scavenge systems.

The H_{ad} characteristic of a scavenge system can be determined on the basis of the following assumptions: The system of passages for the scavenge air and the

Forsch. Ing. Wes., Vol. 8 (1937), p. 57.

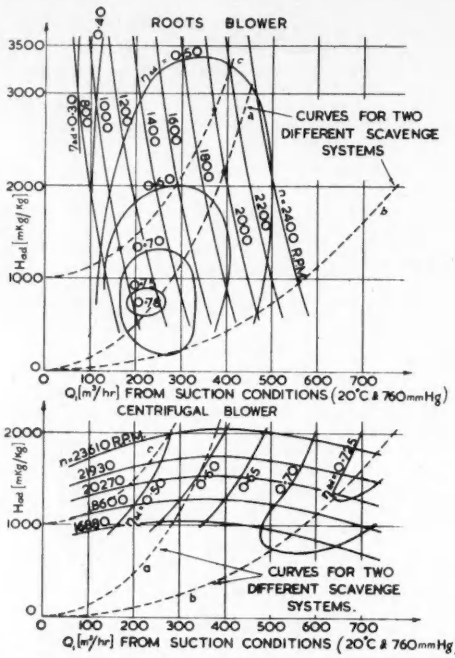
ROOTS BLOWER
CURVES FOR TWO DIFFERENT SCAVENGE SYSTEMS
n=2400 RPM
Q[m³/hr] FROM SUCTION CONDITIONS (20°C & 760mmHg)
CENTRIFUGAL BLOWER
n=23610 RPM
CURVES FOR TWO DIFFERENT SCAVENGE SYSTEMS.
Q[m³/hr] FROM SUCTION CONDITIONS (20°C & 760mmHg)

Fig. 2. **Characteristic curves of a Roots and a radial-vane type blower.**
a, *b* = curves for two scavenge systems, *c* = curve for a scavenge system with increased back pressure in exhaust manifold.

exhaust is replaced by an equivalent nozzle with a variable orifice representing the inlet slots. The air ahead of this nozzle has the state variables of the scavenge air in the air receiver, while the air beyond the nozzle is assigned values corresponding to an adiabatic expansion into the outside air.

With comparatively small exhaust slot areas it is reasonable to assume that the air velocity through these slots is independent of the velocity through the inlet slots, and is only related to the pressure drop between the cylinder and the exhaust manifold. Furthermore, the equivalent nozzle must have a discharge coefficient which is representative of the flow through the cylinder slots. It has been proposed by P. Hold* that for this calculation the area f_{red} of the equivalent nozzle orifice be determined by means of the equation

$$f_{red} = 1 \Big/ \sqrt{\frac{1}{f_e^2} + \frac{1}{f_a^2}}$$

where f_e and f_a are the cross-sectional areas of the inlet and outlet slots respectively of the cylinder. The discharge coefficient μ of the equivalent nozzle is best determined on the scavenge model for various piston positions, from the measured air utilisation factor and the measured pressure drop between the scavenge-air receiver and the outside air. The mean value μ_M of the discharge coefficient is obtained from

$$\mu_M = \int_{+\alpha}^{-\alpha} d\alpha \; \mu \, f_{red} \Big/ \int_{+\alpha}^{-\alpha} d\,\alpha f_{red}$$

$f_{red} = f_{red}\,(\alpha)$ being the cross-sectional area of the equivalent nozzle for a given crank angle α.

For given mean values of the rates of flow, the discharge coefficient increases with increasing dimensions of the scavenge system and, therefore, with increasing Reynolds numbers. It is also slightly affected by the scavenge air pressure.

In high-speed engines it is not possible to obtain a high charge ratio by shifting the beginning of the exhaust slot opening so far forward that the charge pressure at the inlet port opening will already reach the full value required for scavenging. Owing to the shorter exhaust slots used, there will be a partial overflow of residual gases into the air passages, resulting in a reduction of the effective time areas of the inlet slots. The discharge coefficient μ measured on the engine under running conditions will thus be smaller in this case than the discharge coefficient μ_M of the model. This should be taken into account in calculating the slot dimensions. Thus, as soon as the charge pressure is equal to the full scavenge pressure, not only the air volume corresponding to the given air utilisation factor, but also the combustion products which penetrated into the scavenge-air receiver during the preliminary exhaust process, will flow through the inlet slots. The volume of these combustion products, referred to outside conditions and divided by the stroke volume, will be denoted by ΔS.

The time-area integral of the equivalent nozzle orifice, on the basis of the above considerations, is, therefore,

$$\int_0^{\frac{60}{n}} dz\, f_e(z) = (S + \Delta S)\, V_h\, \gamma_a / (\gamma_e\, \mu_M\, w)$$

where z = time (sec), and w = rate of air flow (m/sec) through the equivalent nozzle. The adiabatic delivery H_{ad} which characterises the scavenge system can now be written as follows:—

$$H_{ad} = \frac{(T_e/T_a)}{1960\;\mu_M{}^2\;F^2}\,[60\,n\,V_h\,(S + \Delta S)]^2 \quad .. \quad (11)$$

where n = engine speed (rpm), and $F = \int_{a'}^{-\alpha_e} d\;\alpha\, f_e(\alpha)$.

The adiabatic delivery H_{ad} of the scavenge air blower, therefore, increases as the square of the scavenge air delivered per hour, for given slot dimensions. Thus, for instance, when using a radial vane-type blower mechanically coupled to the engine, assuming the adiabatic efficiency does not vary with the running speed, the same air utilisation factor can be obtained for various speeds. This is only true, however, as long as the velocity triangles of the blower for inlet and outlet are similar.

If the back pressure in the exhaust manifold is greater than the suction pressure of the blower, so that the scavenge system characteristic is similar to curve *c* in Fig. 2, then the drop in the quantity of air delivered per second occurring with decreasing speeds will be greater than that corresponding to the considered speed ratio, and this can sometimes have a marked effect on the air utilisation factor.

With a Roots-type blower, however, the engine behaviour is different at varying speeds. Considering for instance, in Fig. 2, two working points taken as intersections of the Roots blower characteristics for 1000 and 2000 rpm with the curve *a*, representing the scavenge system, it is seen that the ratio of the quantities of air supplied at these speeds is smaller than 2. This circumstance has a favourable effect on the no-load performance of a motor-vehicle engine coupled to a Roots blower, and also makes it possible to operate with a higher back pressure in the exhaust manifold.

The position of the working point in the blower

*MTZ 1941, p. 83.

diagram is also of importance for engine performance, particularly at high piston speeds, because, as shown by eqs. (9) and (11), the power requirement of the blower increases as the cube of the air utilisation factor. If the engine output is calculated by means of eq. (8) for various values of S and various slot dimensions, we obtain the relationship, shown in Fig. 3,

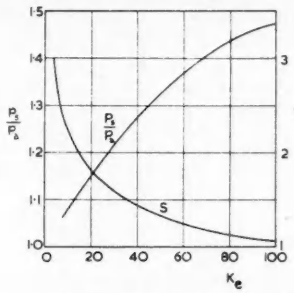

Fig. 3. **Scavenge pressure and air utilisation factor for maximum values of p_e**

between the scavenge pressure and the air utilisation factor S, for various values of the parameter K_e, which is a function of the mean piston speed and the slot dimensions:

$$K_e = c \,/\, [\mu_M \beta_e \zeta \sin \gamma], \quad .. \quad .. \quad (12)$$

where c = mean piston speed (m/sec), β_e = total effective area of inlet slot apertures at right angles to the direction of flow and the periphery of the cylinder, ζ = the stroke/bore ratio, and γ = the inclination of the inlet passages relative to the cylinder axis. This calculated relationship has been confirmed by tests on various scavenge systems. The factor S should thus decrease at reduced piston speeds. If, however, the characteristic of a scavenge system in the blower diagram is such that the air utilisation factor remains practically constant at various speeds, as is the case, for instance, with a radial blower, then the requirement that engine output should be the highest possible at each speed is not fulfilled.

In the upper speed range, the air utilisation factor can only be reduced to the desired value by using special devices, such as inlet baffles, adjustable according to the running speed, for radial blowers. A reduced air utilisation at higher piston speeds also means a less intensive cooling of the piston and cylinder by the scavenge air, and when the value of S is reduced below its normal figure of 1·4–1·5, this effect must also be taken into account.

Piston-type blowers generally make it possible to obtain an almost constant air utilisation factor at various piston speeds, since their air supply ratio is not affected to the same extent by the running speed. In high-speed crankcase-scavenge engines, the influence of the return flow of the combustion products into the crankcase is very considerable at the beginning of the scavenging process, and owing to its dependence on speed, it has a decisive effect on the value of the air utilisation factor.

B. CALCULATION OF THE CHARACTERISTIC PARAMETERS OF A SCAVENGE SYSTEM

High engine performance requires high charge and scavenge ratios. For a high charge ratio, the outlet slots must have minimum heights and the charge pressure must be as high as possible at the beginning of compression. On the other hand, a high scavenge ratio requires a high air utilisation factor for a minimum amount of power for the scavenge blower, that is, the largest possible inlet areas. In order to obtain an optimum adjustment of the values of the parameters involved, it is, therefore, necessary to determine the engine output for various slot dimensions, and then take the dimensions giving the maximum power rating.

For an accurate calculation of the gas-exchange process, the pressure variation during the renewal of the charge can be computed, using step-by-step methods after subdividing the flow process into small time intervals. Provided, however, that the pressure variations in the scavenge air receiver are small, and that wave effects can be avoided in the exhaust system by suitable dimensioning of the exhaust pipes, it is frequently sufficient to assume a mean value for the charge pressure during scavenging, taking, for instance, the arithmetic mean of scavenge pressure and external pressure. In multi-cylinder engines with a common scavenge air receiver, the pressure variations in this receiver can, however, so alter the pressure drop between the scavenge air manifold and the exhaust pipe during the scavenging of each cylinder as to cause substantial changes in the air consumption and power output of the individual cylinders. In such cases, the calculation of the pressure curve also provides a general view of the causes of these differences in output.

In crankcase-scavenged engines, and when piston blowers are used, the scavenge pressure is variable and is also considerably affected by running speed, owing to the return flow from the cylinder to the crankcase occurring at the beginning of the scavenge period. It has not been ascertained by the author whether it is permissible to assume a constant scavenge pressure for this special case, so that under these conditions a detailed pressure calculation is advisable.

The design of a scavenge system can be based on the following considerations:—

(a) *Dimensions of inlet and exhaust slots* : Of the two parameters, the air utilisation factor and the mean scavenge pressure, one can be given an arbitrary value, and the other is then calculated. Usual values for the air utilisation factor are roughly between 0·6 and 0·8 when using crankcase scavenge pumps, and between 1·4 and 1·6 when scavenge air blowers are used.

(b) *Arrangement and positioning of inlet and exhaust slots* : Optimum values are obtained by determining the scavenge ratio for various values of the air utilisation factor. The investigation of flow phenomena in the cylinder is as yet without a theoretical foundation. So far it has only been possible to treat the scavenging process mathematically in the simplest case, a plane expansion type flow with the aid of the potential theory. The scavenge model is, therefore, the best solution for the time being as a means of obtaining reliable data for the design of a scavenge system. It is advisable to use a full-scale model, in view of the fact that the influence of scavenge pressure and the effect of the size and shape of the system upon the flow phenomena are not yet clearly determined.

For a calculation as under sub-section (a) of the air utilisation factor for an assumed height of the inlet slots and a preselected value for the charge pressure p_{ze} at inlet slot opening, it is necessary to know the values of the following quantities:—

1. The sum of the effective slot widths, expressed as a ratio, viz. as the sum of all the inlet or exhaust slot widths perpendicular to the direction of flow, divided by the cylinder periphery, measured in a plane perpendicular to the cylinder axis:

$$\beta_{\text{inlet}} = \beta_e = \Sigma b_e/(\pi D), \; \beta_{\text{exhaust}} = \beta_a = \Sigma b_a/(\pi D).$$

2. The angle γ of the inlet passages [in degrees] relatively to the cylinder axis.
3. The stroke/bore ratio $\zeta = s/D$.
4. The mean piston speed c [m/sec].
5. The effective compression ratio ϵ of the engine.
6. The discharge coefficient of the equivalent nozzle μ_M, and of the exhaust slots μ_a.
7. The pressure ratio p_{za}/p_a = charge pressure at

exhaust opening divided by the external atmospheric pressure.

8. The temperature T_{za} [°C abs.] at the exhaust slot opening, the temperature T_s of the scavenge fluid in the air receiver, the temperature T_e of the scavenge fluid immediately after entering the cylinder, and the suction temperature T_a of the scavenge air.

The following symbols will also be used: σ_e = ratio of effective inlet slot height to stroke, σ_e' = ratio of calculated inlet slot height to stroke, σ_a = ratio of exhaust slot height to stroke, σ' = ratio of piston distance from B.D.C. when $p_z = p_e$, to stroke, w_e, w_a = velocities through inlet and exhaust slots [m/sec], v = specific volume [m³/kg]. The subscript z denotes a prevailing condition inside the cylinder, while the subscript m refers to a condition after expansion in the corresponding control passage.

1. Time-Area Integral of Exhaust Slots.

The height of the exhaust slots, $\sigma_a - \sigma_e$ can be calculated in the usual way for the given expansion ratio from p_{za} to p_{ze}. The time-area integral for the exhaust slots is

$$\int_{z_a}^{z_e} dz f_a = -\int d\left(\frac{1}{v_z}\right) \frac{V_h \rho_a v_{ma}}{\mu_a w_a}$$

where

$$\rho_a = \frac{\epsilon - \sigma_a}{\epsilon - 1} - \frac{\sigma_a + \sigma_e}{2} \quad .. \quad (13)$$

is the ratio of the mean cylinder volume to the stroke volume, during the time interval considered.

Putting

$$\int_{z_a}^{z_e} dz\, f_a = \int_{\alpha_a}^{\alpha_e} \frac{d\alpha}{60\ n} \cdot sD\pi\, \beta_a \sigma_a$$

and

$$K_a = c/(\mu_a \beta_a \zeta) \quad .. \quad .. \quad (14)$$

we obtain, after a few transformations:

(a) for the case of an over-critical pressure ratio occurring during the entire discharge process:

$$p_{ze} \geq 1{\cdot}85\, p_a,$$

$$\frac{\sqrt{T_{za}}}{K_a \rho_a} \int_{\alpha_a}^{\alpha_e} d\alpha\ \sigma_a = 22{\cdot}8 \left[\left(\frac{p_{za}}{p_{ze}}\right)^{0{\cdot}128} - 1\right] = A_1 \quad (15)$$

(b) for the case $p_{ze} < 1{\cdot}85\, p_a$:

$$\int_{\alpha_a}^{\alpha_e} d\alpha\ \sigma_a = -3{\cdot}93 \frac{K_a \rho_a}{K\sqrt{T_{za}}} p_{za}^{r/2} \int_{p_{za}}^{1{\cdot}85 p_a} dp_z\, p_z^{-1-r/2}$$

$$-0{\cdot}706 K_a \frac{p_{za}^{r/2} \rho_a}{p^{1/K}\sqrt{T_{za}}} \int_{1{\cdot}85 p_z}^{p_{ze}} dp_z\, p_z^{-r} [p_z^r - p_a^r]^{-1/2}$$

where $r = (K-1)/K$, and $-1 - r/2 = (1-3K)/(2K)$; the integration gives

$$\frac{\sqrt{T_{za}}}{K_a \rho_a} \int_{\alpha_a}^{\alpha_e} d\alpha\ \sigma_a \simeq 22{\cdot}8 \left[\left(\frac{p_{za}}{1{\cdot}85 p_a}\right)^{0{\cdot}128} - 1\right]$$

$$+3{\cdot}24 \left(\frac{p_{za}}{p_a}\right)^{0{\cdot}128} \left[0{\cdot}924 - 2\psi'^{1/2}\left(\frac{2}{3}\psi' + 1\right)\right]$$

$$= A_2' + A_2'' = A_2 \quad .. \quad .. \quad (16)$$

where $\psi' = (p_{ze}/p_a)^{0{\cdot}256} - 1$.

Fig. 4 shows the variation of A_1 as a function of $1{\cdot}85 p_{za}/p_{ze}$, and the curves for A_2' and A_2'' as functions of p_{za}/p_a for various values of p_{ze}/p_a. The time-area integral required can then be calculated as

$$\int_{\alpha_a}^{\alpha_e} d\alpha\ \sigma_a = A K_a \rho_a / \sqrt{T_{za}}.$$

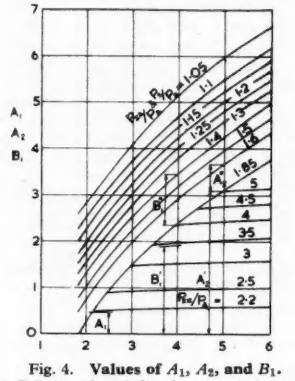

Fig. 4. **Values of A_1, A_2, and B_1.**
A_1, B_1' and B_1'' are plotted for the case $p_{za}/p_a = 4$, $p_{ze}/p_z = 3{\cdot}42$, $1{\cdot}85\ p_{za}/p_{ze} = 2{\cdot}17$, $p_a/p_a = 1{\cdot}4$; while A_2' and A_2'' are plotted for the case $p_{za}/p_a = 5$, $p_{ze}/p_a = 1{\cdot}6$.

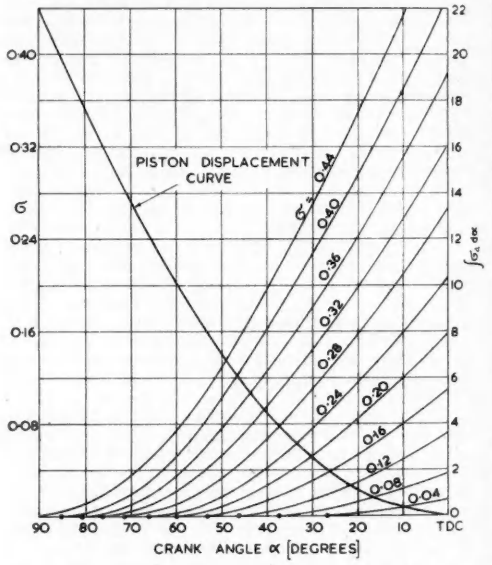

Fig. 5. Time-area integrals for various slot heights from beginning of inlet slot opening to T.D.C.
Ratio of connecting-rod length to crank radius 4 : 1.

Fig. 5 gives time-area values in relation to the crank angle for various slot heights. The horizontal line corresponding to a given value of the time-area integral is followed up to its intersection with the vertical line of the crank angle α_e corresponding to inlet opening, and the required exhaust slot height can then be determined from the parameter of the corresponding curve.

(To be continued)

Problems of Turbine Standardisation

By A. P. Drozdov. (From *Kotloturbostroyenie*, No. 2, 1948, pp. 1-4.)

The article summarises and comments on the standards adopted in the U.S.S.R. for land-type turbo-alternators, listing proposals which were adopted in 1947 and have now become law.

(1) *Turbine Ratings.* For straight condensing and extraction-type turbines, 27 different ratings have been adopted, covering a range of 750 kW to 100,000 kW. In addition, 30 different ratings have been adopted for back-pressure and superimposed turbines with a range of 750 kW to 25,000 kW. Condensing turbines are divided into five groups (Table I*) :—

(a) ten ratings without controlled extraction.
(b) eight ratings with controlled extraction at 1·2 atm. abs.
(c) five ratings with controlled extraction at 5 atm. abs.
(d) one rating with controlled extraction at 7 atm. abs.
(e) three ratings with two controlled extractions.

Standard back-pressure-type turbines will cover a range of back-pressure from 3 to 18 atm. abs. (Table II). It is considered that the total of 57 types should satisfy all needs of electricity supply, industry and district heating installations.

(2) *Steam Conditions.*

(a) The adoption of 90 atm. abs., 480° C for large turbines, as compared with the present 28 atm. abs., 400° C, gives a fuel saving of 15 per cent, and is based on recent achievements of the steel industry. The temperature of 500° C adopted for back-pressure sets is necessary in order to avoid wet steam in the condensing sets on which the back-pressure turbines are superimposed.

When standards are introduced, it is important to leave scope for future development, and it is intended, therefore, to start immediately with the design of experimental turbines with initial steam conditions of 170 atm. abs., 550° C.

The calculated savings for higher conditions compared with 100 atm. abs., 500° C would be as follows :

For 140 atm. abs., 550° C .. 5 per cent
„ 170 atm. abs., 550° C .. 6·5 per cent
„ 170 atm. abs., 600° C .. 8·5 per cent
„ 220 atm. abs., 600° C .. 9·4 per cent
„ 250 atm. abs., 600° C .. 10 per cent
„ 300 atm. abs., 600° C .. 11 per cent

The quality of steel production would permit of building turbines at 170 atm. abs. and 550° C now, and it is envisaged to adopt these as standard conditions in future.

(b) For small turbines the standard conditions are 35 atm. abs., 435° C, yielding a saving of 3 to 6 per cent when compared with the present practice for these ratings, i.e., 29 atm. abs., 400° C.

While it is recognised that higher conditions for turbines of 4000 to 12000 kW would give better efficiencies, the present standards have been agreed in view of practical considerations.

(c) Standardization of vacua is also being aimed at, although this meets with certain difficulties in a country with widely varying climatic conditions, as the vacuum, of course, depends on the cooling water temperature. A 29-inch vacuum at nominal load has been adopted as standard for large machines, wherever the average yearly cooling water temperature does not exceed 10 to 15° C. This vacuum means a saving of up to 2 per cent in the heat consumption of a turbine when compared with a vacuum of 28·5 ins.

(d) The standard back-pressures cover a range of 2 to 17 atm. abs., which should satisfy all district heating and industrial process steam requirements.

For the large number of superimposed turbines

*Certain types of turbines are not reproduced in this table, for their design is not recommended for future manufacture and they should be used only for extensions of existing installations.

TABLE I. Condensing Type Turbines.

Initial Steam Conditions		Turbine Rating	Alternator Speed	Type of Turbine and Extraction Pressure						Feed Water Temperature
				Without Contrld. Extraction	With contrld. extraction at 1·2 atm. abs. for heating	With controlled extraction for process		With 2 controlled extraction pressures		
Pressure	Temperature					at 5 atm. abs.	at 7 atm. abs.	heating at 1·2 atm. abs.	process at 10 atm. abs.	
atm. abs.	°C	kW	rpm		tons/hr	tons/hr	tons/hr	tons/hr	tons/hr	°C
35	435	750	3000 or 1000	—	—	7	—	—	—	150
35	435	1500	3000 1000	×	9	12	—	—	—	150
35	435	2500	3000 or 1000	×	14	18	—	—	—	150
35	435	4000	3000	×	22	25	—	—	—	150
35	435	6000	3000	×	35	40	—	—	—	150
35	435	12000	3000	×	65	—	—	40	50	150
90	480	12000	3000	—	—	—	—	30	40	215
90	480	25000	3000	×	100	—	—	60	80	215
90	480	50000	3000	×	—	—	—	—	—	215
90	480	100000	3000	×	—	—	—	—	—	215

Note : (1) Turbines without controlled extraction marked thus (×).
(2) It is permissible to build extraction turbines for different conditions and extraction quantities, provided that a standard casing can be used.
(3) It is permissible to vary the standard extraction pressure within ± 10 per cent.

TABLE II. BACK PRESSURE TURBINES.

Alternator Output	Alternator Speed	Initial Conditions		Back Pressure atm. abs.					
		Pressure	Temperature						
kW	rpm	atm. abs.	°C	31	18	15	11	6	3
750	1000 or 3000	15	350	—	—	—	—	×	×
1500	1000 or 3000	15	350	—	—	—	—	×	×
2500	1000 or 3000	15	350	—	—	—	—	×	×
750	1000 or 3000	35	435	—	—	—	—	×	×
1500	1000 or 3000	35	435	—	—	×	×	×	×
2500	1000 or 3000	35	435	—	—	×	×	×	×
4000	3000	35	435	—	—	×	×	×	×
6000	3000	35	435	—	—	×	×	×	×
6000	3000	90	500	×	—	—	—	—	—
12000	3000	90	500	×	×	—	×	—	—
25000	6000	90	500	×	×	—	—	—	—

NOTE : Standard Turbines are marked thus (×).
This table includes topping turbines and for that reason covers a range of back pressures above 18 atm. abs.

which will be required in the future in order to improve the efficiency of existing power stations, standard back pressures of 31 and 18 atm. abs. have been adopted.

(3) *Controlled Extraction of Steam.* It is felt that the wide variety of requirements for process steam can well be limited without impairing the genuine needs of the consumers. With double-extraction turbines, where one extraction is normally intended for district heating, it is essential for the district heating planning to fall in line with the adopted turbine standards.

(4) *Turbo-Alternator Speed of Rotation.* Three thousand rpm will be the generally accepted standard for ratings up to 100,000 kW. The alternator speed for geared sets has been chosen as 1000 rpm, in order not to delay present production, although it is realised that this is not the best solution.

SWITZERLAND

The Gas Turbine as a means of Fuel Saving and Increase of Output in Steam Power Stations

By DE REYNIER AND KOWALCZEWSKI. (From *Bulletin Oerlikon*, No. 274, *July/August*, 1948, pp. 1880-1884, 5 illustrations.)

Introduction. Fuel shortage and increasing current consumption put a premium on efforts to increase the efficiency and output of existing steam power stations.

This paper shows that by superimposing an open-cycle gas turbine on an existing steam power plant, considerable cuts in fuel cost can be achieved.

Utilization of Gas Turbine Exhaust Heat. The combustion air for a conventional thermal power station, instead of being led directly to the boiler, is first passed through an air or gas turbine (Figs. 1 and 2). Such combinations enable the utilization of the exhaust heat of the superimposed turbine in the combustion air, with a resulting decrease in heat losses. Owing to the gas turbine, on the air side of the thermal cycle, advantages arise which are comparable to those due to the back-pressure turbine on the steam side, i.e., exhaust heat utilization. If, to achieve higher output, the gas turbine is designed for a larger mass of air than is required for combustion, the heat of the excess air can be given off in a heat exchanger i (Fig. 2), e.g., for heating purposes, or in the economiser k_2 for heating feed water.

If heating of the air is effected through contact with boiler flue gases, the working medium of the superimposed turbine is pure air, which, after expanding in the turbine, utilizes its remaining heat content as combustion air. Such a combined plant could be termed a " steam-air plant " (Fig. 1.)

If, in the superimposed plant, the air is heated by direct combustion of oil in a combustion chamber h, the resulting gas, due to its high percentage of excess air, still represents the combustion air for the boiler of the steam turbine plant (Fig. 2). Both this steam-gas plant, and the steam-air plant superimposed on existing steam turbine plant are suitable means for increasing output without higher consumption of steam and water.

The following considerations apply to a steam-gas plant which can be added to existing thermal plant, while necessitating only minor modifications. Such a combination has the further advantages that the gas turbine plant can be operated independently, is ready at any time, and can, therefore, produce electrical energy before the boiler is fired or when a breakdown of the steam plant occurs.

Example Applied to Gas Turbine. A practical example is presented, and simplified estimates are given :

(a) *The problem.* A factory plant should deliver 1900 kW electrical energy and 8 million kcal/hr heat energy. The latter is utilized for processing down to a minimum temperature of 90° C.

(b) *Solution with Back-Pressure and Condensing Turbines* (Fig. 3). Of the required 1900 kW a back-pressure turbine could deliver 900 kW, and it could also produce the heat energy, the remaining 1000 kW to be produced by a condensing turbine.

The boiler generates 18 ton/hr steam at 25 atmos. abs. and 425° C. The back-pressure turbine operates at a back pressure of 4·5 atmos. abs. and delivers steam to the factory at a temperature of 250° C.

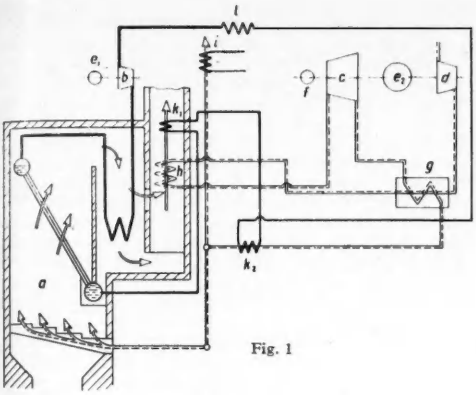

Fig. 1

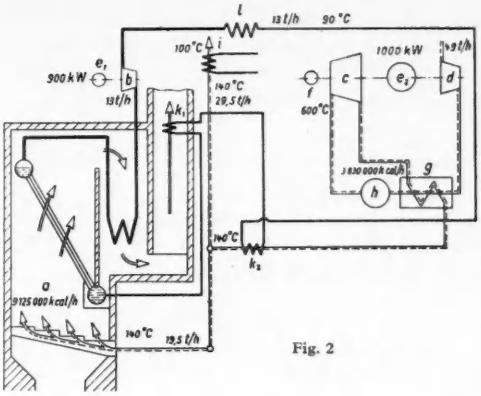

Fig. 2

Fig. 1. **Schematic diagram of a back-pressure and steam-air plant.**

a	= steam generator.	*g*	= heat exchanger (air).
b	= back-pressure turbine.	*h*	= air heater.
c	= air turbine.	*i*	= heat exchanger (water).
d	= compressor.	k_1, k_2	= economiser.
e_1, e_2	= generators.	*l*	= heat consumer.
f	= starting motor.		

Fig. 2. **Schematic diagram of a back-pressure and steam-gas plant.**

a	= steam generator.	*g*	= heat exchanger (gas).
b	= back-pressure turbine.	*h*	= combustion chamber.
c	= gas turbine.	*i*	= heat exchanger (water).
d	= compressor.	k_1, k_2	= economiser.
e_1, e_2	= generators.	*l*	= heat consumer.
f	= starting motor.		

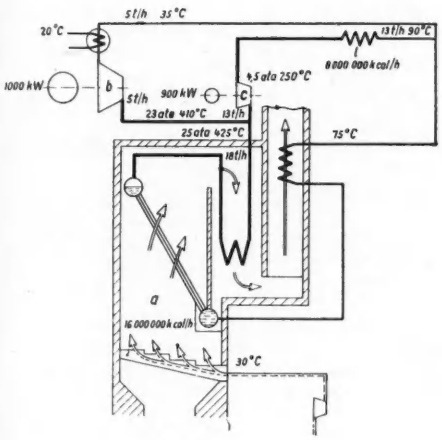

Fig. 3

Fig. 3. **Schematic diagram of a plant according to para. b, with a back-pressure turbine of 900 kW and a condensing turbine of 1900 kW.**

a	= steam generator		gas
b	= condensing turbine group		air
c	= back-pressure turbine group		steam
l	= heat consumer		water

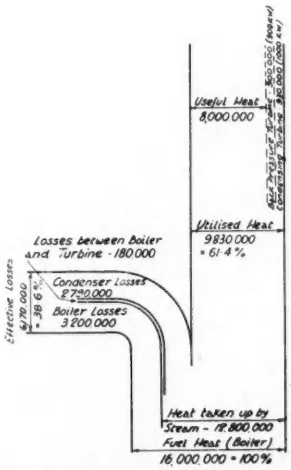

Fig. 4. **Heat-flow diagram of a plant according to para. (b), with a condensing turbine of 1900 kW and useful heat of 8,000,000 kcal/hr.**

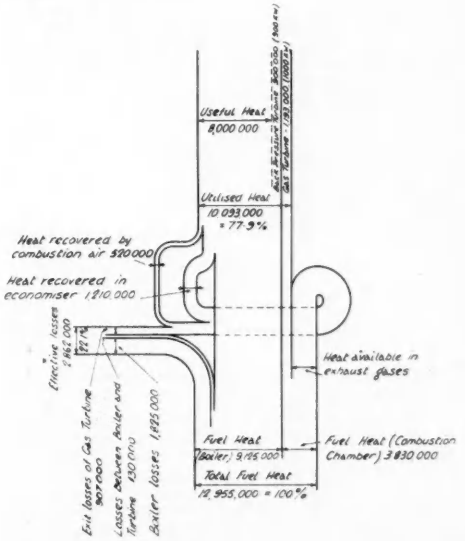
Useful Heat
8 000 000
Utilised Heat
10 093 000
= 77·9 %
Heat recovered by combustion air 520 000
Heat recovered in economiser 1 810 000
Effective losses
1 862 000
14·4 %
Heat available in exhaust gases
Fuel Heat (Boiler) 9 125 000
Fuel Heat (Combustion Chamber) 3 830 000
Total Fuel Heat
12 955 000 = 100 %
Exit losses of Gas Turbine 907 000
Losses between Boiler and Turbine 130 000
Boiler losses 1 825 000

Fig. 5. **Heat-flow diagram of plant according to para. (c), with a gas-turbine of 1900 kW and useful heat of 8,000,000 kcal/hr.**

Estimate of Heat Required from Fuel. The total steam generated is distributed as follows:

Back-pressure turbine	13 ton/hr
Condensing turbine	5 ton/hr
Total steam ..	18 ton/hr

Enthalpy of steam at exit from superheater (25 atmos. abs., 425° C) .. 786 kcal/kg

The mixed condensate (5 ton/hr at 35° C (and 13 ton/hr at 90° C) has a temperature of 75° C at the economiser inlet; enthalpy of same .. .. .. 75 kcal/kg

Heat required from boiler per kg steam 711 kcal/kg

Total enthalpy to be given up to steam :

$$711 \times 18{,}000 = 12{,}800{,}000 \text{ kcal/hr}$$

Required enthalpy of fuel for a boiler efficiency of 80 per cent :

$$\frac{12{,}800{,}000}{0{\cdot}8} = 16{,}000{,}000 \text{ kcal/hr}$$

Boiler losses 20 per cent .. 3,200,000 kcal/hr

Distribution of Heat Given Up to Steam. Between boiler and turbine the pressure decreases to 23 atmos. abs. and the temperature to 410° C, corresponding to a heat loss of about 10 kcal/kg.

	kcal/hr
Total losses between boiler and turbines : 18,000 kg/hr × 10 kcal/kg =	180,000
Heat drop in back-pressure turbine (enthalpy at turbine exit 707 kcal/kg) : 13,000 × (776—707) = ...	900,000
Heat consumption in factory : 13,000 × (707—90) = ...	8,000,000
Heat drop in condensing turbine (turbine efficiency 72·5 per cent) : (776—519) × 72·5 × 5000 = ...	930,000
Heat loss to cooling water (temperature of condensate 35° C) : (593—35) × 5000 = ...	2,790,000
Total heat given up by steam = ..	12,800,000

(c) *Solution with a Gas Turbine Instead of the Condensing Turbine* (see Fig. 2). The condensing turbine is to be replaced by a gas turbine of equal output, resulting in a 5 ton/hr decrease in the quantity of steam required. This decreases the required heat content in the steam as follows :

	kcal/hr
Losses between boiler and turbine : 5000 × 10 kcal/kg = ...	50,000
By eliminating the condensing turbine : 2,790,000 + 930,000 = ...	3,720,000
Total reduction of useful heat = ..	3,770,000

Reduction in fuel heat content :

$$\frac{3{,}770{,}000}{0{\cdot}8} = 4{,}710{,}000$$

The gas turbine for 1000 kW useful power requires approximately 49 ton/hr of air, which is to be heated in the combustion chamber to approximately 600° C. This requires 3,830,000 kcal/hr, of which the following proportion can be recovered :

The exhaust gases leave the heat exchanger at about 240° C and are cooled in the economiser to 140° C; thus 49,000 × 0·247 (240—140) = 1,210,000 kcal/hr are surrendered to the condensate and made available again in the thermal cycle. 19,500 kg/hr air can be utilised as combustion air for the boiler.

It is assumed that, with the condensing turbine, the combustion air has a temperature of 30° C. Owing to higher combustion air temperature (140° C) 19,500 × 0·243 (140—30) = 520,000 kcal/hr of the gas turbine exhaust heat are regained, assuming constant exhaust temperature at the flue exit. The total of the heat recovered is thus 1,210,000 + 520,000 = 1,730,000 kcal/hr.

(d) *Comparison of the Back-Pressure—Condensing Turbine Plant with the Steam-Gas Plant.* To compare the heat consumptions of these two combinations (case b and case c), their heat-flow diagrams, drawn to the same scale, are shown in Figs. 4 and Fig. 5 respectively.

In the following, the efficiencies of these combinations will be compared with the back-pressure turbine.

1. Back-pressure turbine only :

	kcal/hr
Heat consumption of turbine ..	900,000
Useful heat	8,000,000
Losses between boiler and turbine	130,000
Boiler losses 20 per cent	2,270,000
Consumed fuel heat	11,300,000

Power efficiency :

$$\frac{900 \times 860 \times 0{\cdot}8}{900{,}000} = 69 \text{ per cent}$$

2. Back-pressure—condensing turbine :

Extra fuel heat :

$$16{,}000{,}000 - 11{,}300{,}000 = 4{,}700{,}000 \text{ kcal/hr}$$

$$\text{Efficiency} : \frac{1000 \times 860}{4{,}700{,}000} = 18{\cdot}3 \text{ per cent}$$

3. Steam-gas plant :

Extra heat :

$$12{,}955{,}000 - 11{,}300{,}000 = 1{,}655{,}000 \text{ kcal/hr}$$

$$\text{Efficiency} : \frac{1000 \times 860}{1{,}655{,}000} = 52 \text{ per cent}$$

The gas turbine generates power with a 52/18·3 = 2·84 times better efficiency than the condensing turbine.

4. External heat losses :

	With condensing turbine kcal/hr	With gas turbine kcal/hr
Boiler losses	3,200,000	1,825,000
Losses between boiler and turbines ..	180,000	130,000
Condenser losses ..	2,790,000	—
Exhaust gas losses ..	—	907,000
Total losses ..	6,170,000	2,862,000

Reduction of losses with the gas turbine :
6,170,000 — 2,862,000 = 3,308,000 kcal/hr
i.e., approximately 54 per cent.

5. Consumed fuel heat :

	kcal/hr
With the condensing turbine (Fig. 4)	16,000,000
With the gas turbine (Fig. 5) ..	12,955,000
Saving	3,045,000

i.e., approximately 19 per cent.

A 1,000 kW gas turbine plant was in uninterrupted service for several months at the Oerlikon works, and during that time has supplied over one million kWh to the grid of the city of Zurich.

The utilization of the gas turbine exhaust gases as combustion air for the steam boiler up to a temperature of 200° C proved quite practicable.

New Balancing Machines for the Dynamic Balancing of Rotors

By E. LISELIUS. (From *Ericsson Review*, No. 3, 1948, pp. 87-100, 13 illustrations.)

An unbalanced rotor is a body which revolves about a supporting axis which does not coincide with the principal axis of inertia nearest to it.

The body is statically unbalanced if the supporting axis does not pass through the centre of gravity of the rotor. The centrifugal force then generated acts at the centre of gravity, and its magnitude is given by

$$P = m\omega^2 e \qquad .. \qquad .. \qquad (1)$$

where m = mass of body, ω = angular velocity, and e = distance between the centre of gravity and the supporting axis.

Furthermore, the rotor will be dynamically unbalanced if the inclination φ of the supporting axis relative to the principal axis of inertia nearest to it is not zero. This generates a couple

$$M \simeq (I_x - I_z)\,\varphi\omega^2 \qquad .. \qquad .. \qquad (2)$$

where I_z = moment of inertia of rotor about the principal axis Oz closest to the axis of rotation, and I_x = moment of inertia of rotor about the principal axis Ox at right angles to the Oz axis.

STATIC LACK OF BALANCE

When a statically unbalanced rotor rotates at a speed above the critical whirling speed, it tends to rotate about a centroidal axis, known as the *free axis*, parallel to the axis of rotation of the rotor body. Consequently, if the bearings supporting the rotor can oscillate freely, their centres will describe circles as the body rotates, and both bearings will deflect in the same direction at any instant. With rigidly-mounted bearings, when the rotor runs at a speed below the critical, undesirable bearing forces are set up by the unbalanced force, and give rise to considerable vibration.

DYNAMIC LACK OF BALANCE

Dynamic unbalance, apparent only when the body is rotating, may be observed if we try to balance a rotor disc with an unbalanced mass m_1 by adding a second disc weighted with a mass m_2 with a phase-angle $\alpha = 180$ degrees relatively to m_1. If $r_1 m_1 = r_2 m_2$ the rotor will be statically, but not dynamically, balanced (see Fig. 1). Provided the bearings do not restrain its motion, the rotor will not rotate about the supporting axis but about the axis A', the free axis or principal axis of inertia passing through the centres of gravity of the two discs, and forming an angle φ with the supporting axis. The shaft axis then rotates over the surfaces of two cones having their apexes at the centre of gravity O of the rotor system.

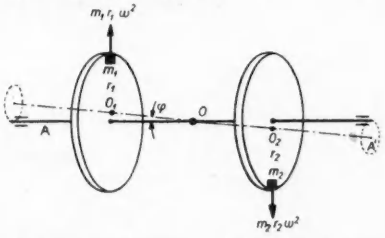

Fig. 1. **The rotor is balanced statically when $r_1 m_1 = r_2 m_2$ and the angle between m_1 and m_2 is 180°, but dynamic lack of balance will occur when the rotor is revolving.**

In the most general case, the rotor is unbalanced both statically and dynamically, that is, we have $r_1 m_1 \neq r_2 m_2$ and $\alpha\ (m_1, m_2) \neq 180°$. The free axis forms an angle φ with the supporting axis of the system, provided that the angle α between m_1 and m_2 differs from 180 degrees without intersecting the shaft axis. The shaft axis then describes a hyperboloid while rotating about the free axis.

HOW IS A ROTOR BALANCED ?

Considering Fig. 2, let P be the force due to static unbalance, and $M = Kl$ the torque produced by dynamic unbalance. If it can be shown that the force P applied at the centre of gravity, and the couple Kl, acting about the shaft axis, can be replaced by two forces R_A and R_B applied in different planes at arbitrarily selected points, it is obvious that by applying equal but oppositely-directed forces $-R_A$, $-R_B$, it will be possible to counteract R_A, R_B, and therefore P and the couple Kl. These additional forces may be obtained as the centrifugal forces of two masses applied in the planes containing points A and B, and perpendicular to the rotor axis.

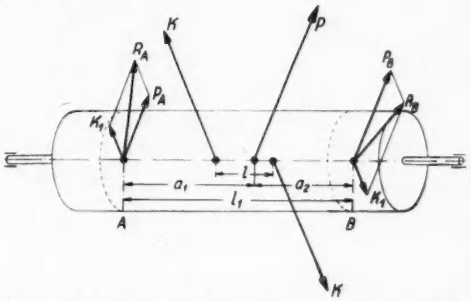

Fig. 2. **The force P, acting at the centre of gravity, and the couple K, can both be replaced by the forces R_A and R_B in the balancing planes.**

The couple Kl can be replaced by an equivalent couple $K_1 l_{AB} = Kl$, where l_{AB} is the axial distance between the points A and B, K_1 being parallel to K. Furthermore P is resolved into two forces P_A and P_B acting at points A and B, and determined by the two conditions $a_1 P_A = a_2 P_B$, and $P_A + P_B = P$, so that

$$P_A = a_2 P/(a_1 + a_2), \text{ and } P_B = a_1 P/(a_1 + a_2).$$

The forces R_A and R_B are then obtained as resultants of the vectors P_A, K_1, and P_B, K_1.

In general, it is preferable to refer the lack of balance to the particular planes in which balancing is to be carried out, and it may then be assumed that the lack of balance in the subsequently balanced rotor was due to the introduction of an out-of-balance force in each of these planes.

In a modern balancing machine, the measurement of the static and dynamic unbalance components is carried out using two different methods: (1) the body is supported on springs or resiliently mounted bearings, so that it can oscillate freely, or (2) it is supported at a point of oscillation while the cradle or resiliently supported rotor bearings may be fixed at one of two selected points. With a freely oscillating rotor the body will rotate about an axis which is very close to the principal axis of inertia, since the inertia forces will be in equilibrium with the spring forces which are very small.

Two measuring points for determining the oscillation amplitude of the supporting axis, taken for instance immediately in front of the bearings, will suffice to determine the positions of the supporting axis and the free axis of rotation in relation to one another.

The measurement points, however, are only in

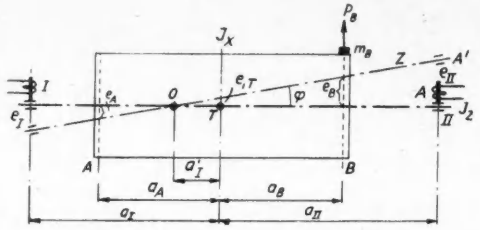

Fig. 3. **The distance between the centre of gravity T and the centre of oscillation O of the rotor depends on the geometry and mass of the rotor and the distance of the balancing plane from the centre of gravity. It is, therefore, a characteristic parameter for each given type of rotor.**

exceptional cases situated in the balancing planes, and consequently, with a freely oscillating rotor it is necessary to find a relation between the amplitudes e_A and e_B in the balancing planes, and e_I and e_{II} at the measuring points.

In order to find this relation, we consider a perfectly balanced rotor having the mass m in the plane B, which is loaded with an unbalance mass m_B producing a force P_B (see Fig. 3). The axis of rotation will then be displaced until its distance e_{1T} from the centre of gravity, and its angle φ with regard to the supporting axis, assume such values that dynamic equilibrium is established. The force is then (assuming $\omega = 1$ rad/sec to avoid the repetition of ω^2 in the following calculation):

$$P_B = m\, e_{1T} \quad .. \quad .. \quad (3)$$

and taking the moments with regard to the assumed centre of oscillation, O, gives

$$P_B\,(a_B + a_I') = (I_x - I_z)\,\varphi + m\, e_{1T}\, a_I' \quad .. \quad (4)$$

where I_x = moment of inertia about a transverse axis passing through the centre of gravity T, while I_z = moment of inertia about the new axis of rotation A'.

From eqs. (3) and (4) we have

$$m\, e_{1T}\,(a_B + a_I') = (I_x - I_z)\,\varphi + m\, e_{1T}\, a_I'$$

so that $\quad e_{1T}/\varphi = (I_x - I_z)/(m\, a_B)$

From Fig. 3, $\varphi = e_{1T}/a_I'$ or $a_I' = e_{1T}/\varphi$, and, therefore,

$$a_I' = (I_x - I_z)/(m a_B) \quad .. \quad .. \quad (5)$$

Therefore, the distance a_I' of the centre of oscillation to the centre of gravity is independent of P_B. The value of a_I' is only affected by the geometrical mass characteristics of the rotating body, and by the balancing plane distance to the centre of gravity, a_B; thus, a_I' is a characteristic parameter for any given type of rotor. Furthermore, eqs. (3) and (5) give

$$P_B = m\, a_I'\, \varphi$$

and using the relations between e_I, or e_{II}, and φ, we obtain

$$P_B = -\,m\, a_I'\, e_I/(a_I + a_I'),$$

$$\text{or } e_I = -\,(a_I + a_I')\, P_B/(m a_I')$$

and

$$P_B = m a_I'\, e_{II}/(a_I - a_I')$$

$$\text{or } e_{II} = +\,(a_I - a_I')\, P_B/(m a_I')$$

The amplitude of the oscillation set up by P_B in the plane of the measuring point II is thus $(a_I' - a_I)/(a_I + a_I')$ of the amplitude in the balancing plane I. If, therefore, a corresponding part of the amplitude e_I at I is added to e_{II} at II, the resultant value obtained will be independent of P_B.

If the balancing machine is of the freely-oscillating type on resiliently mounted bearings, the oscillating points are shifted to the planes in which corrections are to be undertaken. When the cradle is fixed in one of the balancing planes, the unbalanced component at this point has no influence on the oscillating movement of the cradle, and the rotary forces cannot cause any movement at this fixed point since the lever arm is zero. The body will rotate about an axis determined by the condition that the sum of all the moments acting on the point of oscillation is zero.

In practice, the required corrections are not calculated from the rotor data I_x, I_z, m, but the machine is calibrated by placing known correction weights on the actual body to be balanced, and the oscillations then observed indicate the relation between the unbalanced moments and the measured values.

MEASURE OF THE MAGNITUDE OF THE UNBALANCED EFFECT

If the out-of-balance weight q is situated at a radial distance r from the shaft axis, then the unbalanced moment is

$$S = rq$$

and the unbalanced force is $R_A = q\omega^2\, r/g$. The degree to which a rotor must be balanced is directly proportional to the rotor mass. For heavy rotors, a greater unbalanced effect is permissible than for light ones under otherwise similar conditions. To indicate the relative accuracy of balancing, a relative efficiency figure can be defined as follows:—

$$\rho = qr/Q$$

where Q = weight of the rotor. This efficiency figure ρ is a measure of the radial displacement of the centre of gravity, and is useful in determining the sensitivity of a balancing apparatus.

BALANCING APPARATUS TYPE YRB 10 FOR SMALL BODIES ROTATING AT HIGH SPEEDS

The balancing apparatus type YRB 10 is shown in Fig. 4, and, schematically, in Fig. 5. The rotor shaft being tested is mounted vertically in a ring supported radially by three horizontal leaf springs. The rotor then performs small constant-amplitude vibrations about the point of intersection of the springs with the stationary rotor axis. As one of the balancing planes of the rotor coincides with the plane of the springs, the shaft ends of the unbalanced rotor will describe circles while the rotor rotates.

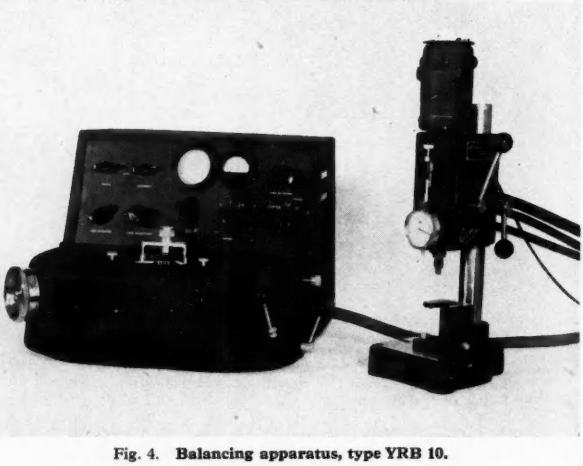

Fig. 4. **Balancing apparatus, type YRB 10.**

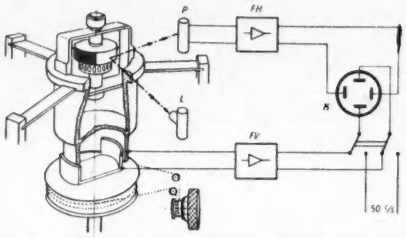

Fig. 5. **Diagrammatic view illustrating the principle of the balancing apparatus type YRB10.**

FH = photocell amplifier.
FV = main amplifier.
K = cathode ray tube.
L = lamp.
P = photo-cell.

Vibration pick-up:—The lower part of the suspension ring is a hollow cylinder which also performs a similar movement. The stand under this cylinder carries a metal plate with two small circular segments diametrically arranged in such a way that they can rotate inside and outside the hollow cylinder. The cylinder and segments thus form a cylindrical type condenser with an air-gap varying in accordance with the shaft vibration. When the condenser is charged with a d.c. voltage, the voltage between the segments and the sleeve will vary with the changes in capacitance, and these voltage variations are amplified and indicated as vertical deflections of the beam of a cathode ray tube. The amount of unbalance is thus determined by the amplitude of the variable a.c. voltage on the oscillograph screen, and this voltage is also measured by a voltmeter graduated in millimetres drilling depth, which indicates directly the quantity of material to be removed from the rotor at a given distance from the centre.

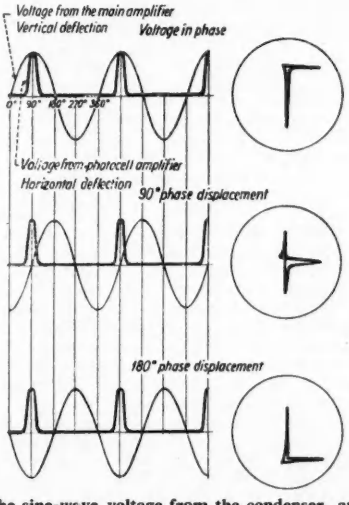

Fig. 6. **The sine-wave voltage from the condenser, and the reference voltage from the photo-cell (this latter voltage being converted into short positive pulses after passing through the impulse generator), produce characteristic figure on the oscillograph screen.**

The horizontal deflection of the trace is obtained by illuminating the rotor periphery by means of a lamp. Part of the periphery has a dull finish, while the other part is polished so that a light pulse will be reflected on to a photo-cell at each revolution. The photo-cell current is then amplified and used as a timing pulse for the cathode-ray tube.

The condenser and photo-cell voltages produce characteristic figures on the oscillograph screen; the size of these figures depends on the unbalance amplitude, while the characteristic shape depends on the phase position of the condenser voltage in relation to the photocell voltage. Thus, the phase position of the condenser is used to determine the position of the " heavy section."

To avoid having two sinusoidal voltages on the plates of the cathode-ray tube, producing elliptical figures on the screen which are not suitable for indication, the photo-cell voltage is converted into short, positive pulses in an impulse transmitter and a more suitable indication figure is thus obtained. Fig. 6 shows examples of the Γ, ⊢, and L type figures obtained on the oscillograph screen corresponding to in-phase, 90° and 180° out-of-phase voltages. The photo-cell voltage is used as reference voltage, since the photo-cell and the light source are fixed. The phase position of the condenser voltage is varied by rotating the condenser segments around the cylinder, by means of a knob graduated for 360 degrees. The knob is rotated manually until an L-shaped figure is produced on the screen, and the number of degrees shown at the knob will then indicate the angle between the heavy section and the dull/polished transition point of the rotor.

After stopping the rotor, it is set up together with its frame in a drilling machine fixture, and the point requiring machining is then determined using a graduated disc laid over the rotor in such a way that the figure for the degree reading comes immediately under the drill. The drilling machine motor is then started by moving an arm on the machine, and at the same time a valve is opened in an ejector which draws off the chips. The depth of the drilled hole, which has been indicated by the voltmeter, is read off on an indicator. When this plane has been balanced in the manner described, the rotor and frame are turned and the entire process is repeated for the other balancing plane.

A band-pass filter is included in the electrical circuit of the balancing machine to eliminate high-frequency vibrations due to defective bearings. The condition of the bearings can also be checked after the rotor has been balanced. For this purpose the main amplifier is disconnected and any vertical deflection on the oscillograph screen may be traced to defective bearings. A second filter is built in the photo-cell amplifier to compensate phase displacement due to the main amplifier filter, and also to cut out spurious voltages which may be set up as a result of any slight inaccuracies in the production of the dull and polished surfaces on the rotor. The running speed is determined by stroboscopic means, using the 50-cycle mains supply voltage as a vertical time sweep. The balancing machine is normally run at 9,000 rpm and, after switching-in the speed control circuit, this speed is reached when an E-shaped figure is produced on the cathode-ray tube screen.

The main amplifier transmits the signal voltage from the condenser type pick-up to the grid of a triode-heptode ECH 21, of which only the heptode portion is employed, the triode being earthed. The valve output is then fed back through an RC-filter to the injector grid. After amplification in a second valve, type EF 22, the condenser voltage is led via a change-over switch to the vertical plates of the cathode-ray tube or to the volt-meter.

The photo-cell amplifier transmits the voltage due to the photo-cell current flowing through the input resistance to the grid of the first valve, also of type ECH 21, and the grid bias frequency thus varies with the rotor speed. This valve also has a negative feedback. From the second valve, type EF 22, the amplified voltage is led to the third valve, type 1852, a pentode with a steep characteristic. This valve has a negative bias of approximately — 25 volts, and, consequently, the

valve is blocked so that only the peak of the positive half-cycle can pass through to the anode circuit. The short pulses thus obtained are led via a condenser to the horizontal plates of the cathode-ray tube.

This apparatus is specially designed for measuring the lack of balance in rotors of aircraft gyroscopes, but it can also be used for other high-speed rotors with a diameter not exceeding 60 mm. The apparatus is fitted with two air nozzles for driving the rotor up to the desired speed, one for each direction of rotation. Electrically driven rotors can also be tested.

BALANCING MACHINE, TYPE YRB 50

This machine, shown in Figs. 7, and 8, is of the freely-oscillating rotor type previously mentioned. The rotor is mounted in two open bearings driven through a link-type coupling permitting transverse vibrations of the rotor shaft. The bearings are elastically mounted so as to be able to perform small horizontal vibrations of substantially constant amplitude, and each bearing is connected to a vibration pick-up of the moving-coil type.

The bearing vibrations of the unbalanced rotor running at a constant speed are converted into sinusoidal voltages by the moving-coil pick-ups. Either one of these voltages can be transmitted to a coil of a watt-meter. The other watt-meter coil is fed from one of two generators driven by the main motor, and the constant voltage thus generated will serve as a reference voltage, with the same frequency as the voltage from the pick-up. With suitable phase adjustment, the instrument deflection will then be proportional to the difference voltage, which depends on the magnitude of the lack of balance. The phase position of the generator stators can be varied manually, and the number of degrees of phase adjustment is indicated by a pointer on a graduated scale. The angular position of the unbalanced section of the rotor is determined by the corresponding setting of the generator stator. According to whether the instrument deflection is to the right or the left of its zero position, the out-of-balance section must be compensated by removing or by adding material. An automatic position adjustment system can also be used.

As previously stated, the forces R_A and R_B are eliminated by adding or removing material in the two balancing planes, thereby altering the position of the centre of gravity in each of these planes. A force added in one balancing plane of a freely oscillating rotor will, however, affect both balancing planes and therefore the vibration amplitude of both bearings. Thus, for a rapid measurement of the forces to be compensated in each of the balancing planes, it is necessary to introduce a correcting system by which the additional forces are corrected automatically. In the machine being described, the correction is achieved electrically by means of a correcting network.

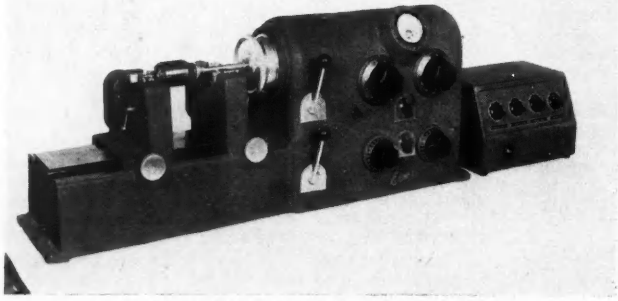

Fig. 7. **Balancing machine, type YRB 50**

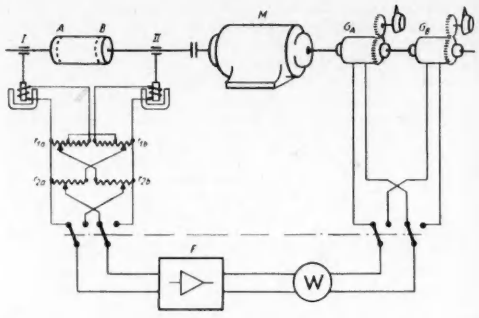

Fig. 8. **Diagrammatic view showing the principle of the balancing machine, type YRB 50.**

I, II = measuring points.
A, B = balancing planes of rotor.
G_A, G_B = reference-voltage generators for each of the balancing planes.
r_{1a}, r_{1b}, and r_{2a}, r_{2b} = rheostats in the correcting network.

The voltages from the vibration pick-ups are taken out through two potentiometers r_{1a} and r_{1b} (Fig. 8) between the central tapping and a sliding contact which can be adjusted so that either a positive or a negative part of the voltage generated by one pick-up is added to that of the other; the resultant voltage is then proportional to the lack of balance in the plane of measurement. The potentiometer adjustments for a given rotor type are a function of the distance of the measuring point and correction plane from the centre of gravity, and usually differ on each side of the latter. The resistances r_{2a} and r_{2b} produce the deflection corresponding to the lack of balance being determined, and the instrument can be calibrated in such a way that its deflection will indicate the particular quantity in which the out-of-balance correction is being measured, i.e. grammes, millimetres drilling depth, etc. All the rheostats are operated by a handwheel on the front of the amplifier, and have given settings for each particular type of rotor.

The pick-up voltages pass through the correcting network to the amplifiers where they are amplified in three stages before going to the watt-meter. The three amplifier valves are type 6J7 pentodes, the last valve, however, being used as a simple triode. The potentiometer for amplification control is in the grid circuit of the second valve. As the watt-meter only gives a deflection when the currents in the two windings are of the same frequency, it is not necessary to include a filter in the amplifier circuit in order to cut out higher-frequency voltages due to ball bearing rattle, external vibrations, etc.; moreover, the harmonics of the reference voltages are very low.

This type of machine has been designed in two sizes, type YRB 50 for bodies weighing from 0·1 to 5 kg, and type YRB 60 for rotors between 3 and 100 kg. Further details of the three Ericsson machines described in this article are given in the following table:—

Machine Type.	YRB 10	YRB 50	YRB 60
Distance between bearings (mm)	20–50	35–300	70–1000
Max. shaft diameter (mm) ..	—	35	70
Max. rotor diameter (mm) ..	60	270	600
Weight of rotor (kg)	0·05–1·0	0·1–5·0	3–100
Balancing speed (rpm) ..	9000	1800	1200
Sensitivity (minimum vibration amplitude, microns) ..	0·03	2·0	5·0
Motor power (hp)	—	0·25	1·5

New Facts on the Straightening of Welded Aircraft Structures

By A. Y. Brodsky. (From *Avtogennoye Delo*, No. 8, 1947, pp. 14-19, 10 illustrations.)

A large number of aircraft structures in which the most important components are essentially rods or tubes are built-up by welding. Among other requirements, interchangeability and strength are very important.

The welding process necessitates local heating, which normally carries plastic deformation and warping of the structure and this must, therefore, be considered an inherent feature of the process. Bending of the structural elements, out of their geometric axes, reduces both the degree of interchangeability and strength.

Therefore, straightening is necessary to improve these characteristics, but it must obviously be accompanied by a whole series of other technological processes.

(1) *The uniform heating of a rod under different conditions of clamping and stressing.*

The properties of steel change with temperature, but for the sake of simplicity, it is assumed that, for a steel with small carbon content, the yield point remains constant up to 500° C and then falls to zero at 600° C.

For Chromansil—the steel used for most welded aircraft structures in the U.S.S.R.—these temperature limits are not known, but they may be designated T_I and T_{II}. Analysis of diagrams of stresses and deformations for uniform heating and cooling and for different methods of clamping reveals that:—

(a) With the uniform heating of a rod having a limited expansion in the transverse direction, and freedom of longitudinal contraction there will be no residual stress or deformation if $(\zeta - \lambda) < \epsilon_s$ where ζ is the relative elongation, λ is the actual elongation, and ϵ_s is the relative elongation which is not realized, corresponding to the yield point. If $(\zeta - \lambda) > \epsilon_s$ the rod will contract to the limit of maximum plastic deformation when heated.

(b) When the ends of the rod are clamped, heating above T_{II} is not reflected in the residual deformation and stress.

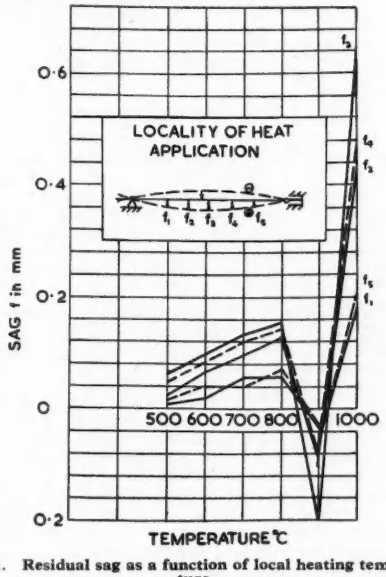

Fig. 1. **Residual sag as a function of local heating temperature.**

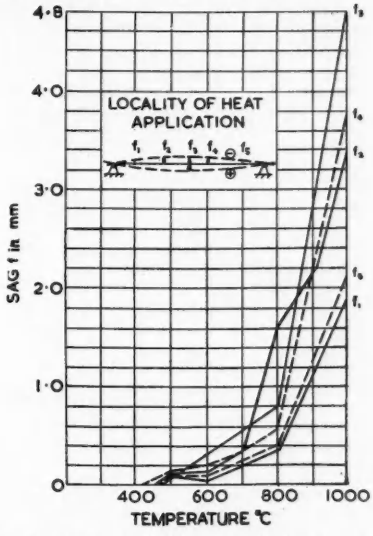

Fig. 2. **Residual sag as in Fig. 1, but with freedom of angular movement only.**

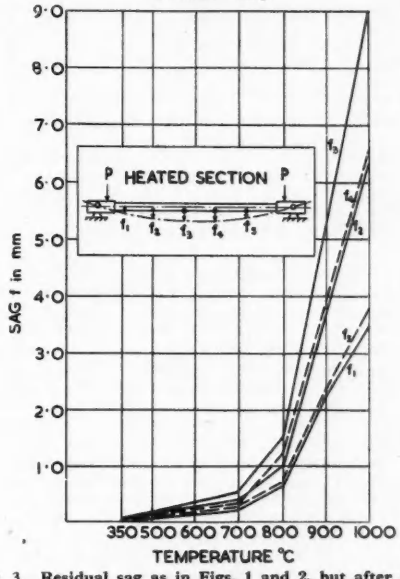

Fig. 3. **Residual sag as in Figs. 1 and 2, but after initial elastic deformation which is not fixed.**

(c) A fixed initial elastic deformation is equivalent to heating and does not affect the residual stresses. Ageing at high temperature also does not affect residual deformations or stresses.

(d) When the initial elastic deformation is not fixed, ageing affects the residual deformation of the rod.

If the effects of locally heating a section of a member of tubular cross-section are examined, it

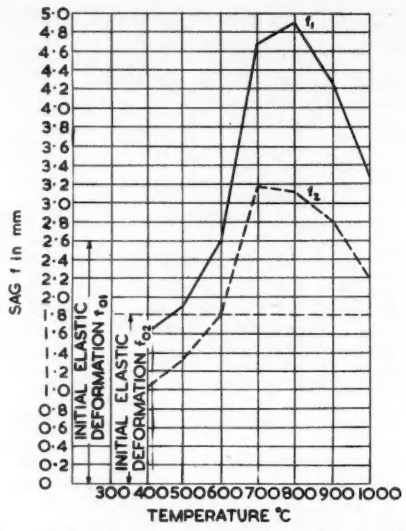

Fig. 4. **Residual sag as in previous figures, but after initial fixed elastic deformation.**

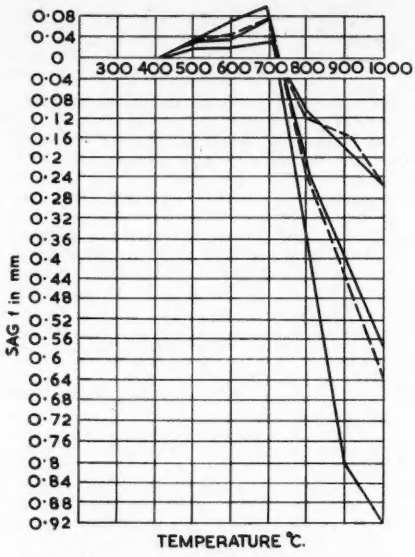

Fig. 5. **Residual sag, unlimited freedom of movement, rapid cooling with the aid of liquid coolant.**

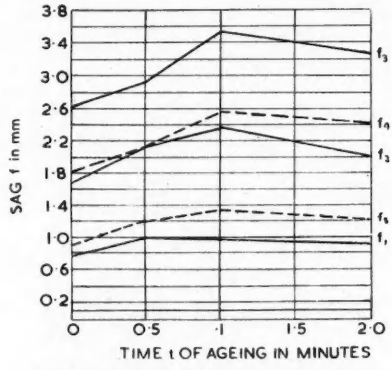

Fig. 6. **Influence of ageing on residual sag.**

can be assumed that the heated section represents a uniformly heated rod and it can be stated to all intents and purposes that the deductions arrived at for the uniform heating of a rod will equally apply.

(2) *Effect of temperature when straightening by local heating.*

The optimum temperature and the values of T_I and T_{II} have been established experimentally with tubes of Chromansil steel.

The principal requirements for straightening by local heating were established by using an originally straight specimen. Special multi-nozzle type burners were designed, which ensured stable heating conditions for the experiments. Figs. 2 and 3 represent the results obtained with different clamping methods and give the residual sag as a function of the temperature.

The following deductions can be made from Figs. 2 and 3.

(a) Beginning with a temperature of 350-450° C, samples show a residual sag steadily increasing with the temperature. From 800° C the absolute values of residual sag show a rapid and staggered increase up to a maximum at 1000° C.

(b) If there is complete freedom of movement of the ends of the sample the sag changes its sign at 900° C, this being due to an allotropic change of α-iron to γ-iron. As the freedom of movement of the ends of the samples is more and more restrained so the physical processes are correspondingly restricted. Fig. 4 shows the residual sag obtained when straightening by local heating, and when an elastic deformation has been initially applied. The curves are similar to those in Fig. 3, but their peak occurs at 700° C, indicating that an elastic deformation is equivalent to heating.

In order to obtain data on the importance of the rate of cooling, liquid cooling was applied to some samples, the results being shown in Fig. 5. It will be seen that at 700° C the sign of residual sag changes and remains the same up to 1000° C. The change of sign at 700° C occurs because the pearlitic structure of the pre-eutectic steels changes into austenite. On rapid cooling from 850° C to 700° C, partial hardening sets in and the austenite changes into martensite. The specific volume of steel depends on its micro-structure; consequently, instead of contraction, expansion occurs in the course of the cooling process.

The results of these experiments can be summarized:

(a) Maximum residual sag occurs when heating is carried on to 1000° C.

(b) Initial elastic deformation causes the maximum sag to occur at 700° C.

(c) Higher rate of cooling than the normal (i.e. in still air) causes the sign of sag to change.

The optimum temperature for the straightening of parts by local heating is 800-850° C. Cooling should be carried out as slowly as possible.

(3) *Efficiency of straightening in a rigid clamping device.*

Figs. 1, 2, and 3 offer comparisons of residual sags for different methods of clamping. When there is unlimited freedom of movement the maximum sag is 0·7 mm (Fig. 1); when axial movement is limited the sag becomes 4·8 mm (Fig. 2); and when angular movement is limited the sag increases to 9·0 mm (Fig. 3). This proves the desirability of rigid fixing of the members when they are being heated.

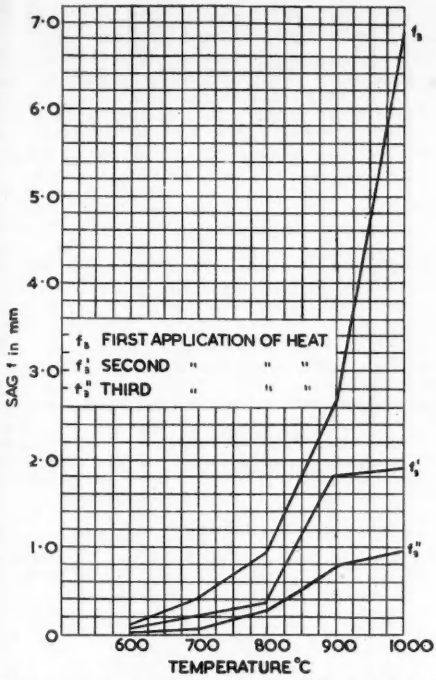

Fig. 7. **Influence of repeated local application of heat.**

(4) *Effects of ageing at high temperature and of repeated heating on the straightening of parts.*

Fig. 6 gives the sag as a function of the period of ageing; after an initial increase the sag decreases. The initial increase is due to the longer period during which the heat affects the metal, and the decrease is due to a levelling out of the temperature through the whole length of the rod. Ageing does not appear to be economical.

Fig. 7 shows the influence of repeated heating on the sag; while the total residual sag increases with each heating the rate of increase is gradually reduced.

(5) *Relation between residual sag and the sag due to initial elastic deformation, when straightening is carried out in the cold state.*

This relation was established as shown in Fig. 8. The equation of the curve is

$$f_1 = f_2/(1 - D \operatorname{cotan} \alpha) + f_s$$

where f_1 is the sag due to the initial elastic deformation in mm, f_2 is the residual sag in mm, α is the angle

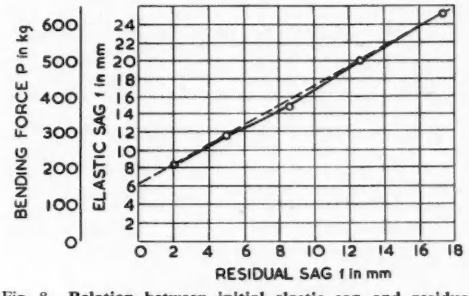

Fig. 8. **Relation between initial elastic sag and residual sag when the cold-straightening process is applied.**

of the deflection curve, and f_s is the sag corresponding to the yield point in the most highly stressed layer of the rod.

(6) *Straightening of hardened elements.*

Straightening of hardened parts leads to a reduction of all their mechanical properties and, therefore, must not be attempted.

AUSTRIA

Electrical Measurement of Acceleration

By K. MELZER. From *Elektrotechnik und Maschinenbau*, Vol. 65, No. 12, December, 1948, pp. 203-208, 6 illustrations.)

ACCELEROMETERS of the seismic-mass type can only be used over a certain range of frequencies. In particular, when mounted in motor vehicles, they do not give fully satisfactory indications during slowly-varying processes such as occur during the starting and running down periods. Furthermore, these accelerometers are sensitive to the gradient of the road or railway along which the vehicle is travelling.

On the other hand, electrical accelerometers have also been developed in the form of velocity-meters with an electrical differentiating circuit. For instance, angular acceleration can be measured by recording the charging current of a condenser to which is applied a voltage proportional to velocity, such as is produced by a d.c. tacho-generator, with the condenser in its armature circuit, the current variation being indicated by a galvanometer-type oscillograph. The readings are, however, affected by irregularities in the commutation and the brush gear, and by the leakage currents through the condenser.

These difficulties can be avoided by using an a.c. machine of the inductor type, without slip rings, as a speed-sensing device. The a.c. voltage is rectified and then goes to the measuring condenser. The rectified pulsating voltage then comprises a voltage proportional to acceleration, and harmonics which may affect the instrument indications at low operating speeds. The a.c. machine used as a tacho-generator must, therefore, be designed with as many poles as possible, so that the charging current will have a higher frequency. The harmonics are then no longer dangerous, but can be reduced to any desired degree by means of a smoothing condenser. The error due to the leakage current is compensated by an auxiliary current proportional to the instantaneous charging voltage.

The circuit diagram of an acceleration-measuring system with leakage-current compensation is shown in Fig. 1. The measuring circuit consists of the a.c. machine serving as a pick-up, the rectifier G, the measuring condenser C_m, the ohmic resistance R, the smoothing condenser C, and the indicating instrument R_m. In parallel with the measuring circuit is the auxiliary half-wave rectifier G_k, the smoothing condenser C_k, the voltage divider P_k, and the resistance R_k, in which the compensation current is generated. Provided the value of R_k is chosen sufficiently high,

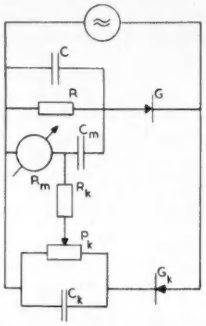

Fig. 1. **Schematic diagram of the accelerometer circuit with leakage-current compensation.**

the compensating circuit will have no effect on the sensitivity and the time constant of the measuring circuit.

The sensitivity of the circuit can be determined from the capacity of the condenser. Since, however, this is not generally known with sufficient accuracy, but changes with ageing or other influences, a direct calibration with a known time-varying alternating voltage is to be preferred. The reference voltage variation must be continuous, so that a tapping from a slide resistance cannot be used. The best solution is a voltage divider with a liquid resistance.

The voltage divider comprises a copper rod dipping into a glass tube containing a copper sulphate solution, with electrodes at both its ends, a known alternating voltage with a constant peak amplitude being applied to the two electrodes. The rod is moved into and out of the solution with a known adjustable velocity. From tappings at one of the electrode terminals and the flexible wire connected to the moving copper rod, a time-varying voltage is obtained, and this can then be introduced into the measuring circuit instead of the a.c. voltage from the tacho-generator. The voltage variation in volt/sec can also be expressed in rev./sec² if the relation between the e.m.f. and the tachometer rpm is known. In vehicle applications, the e.m.f. depends on velocity and, therefore, the voltage variation can be expressed as an acceleration in m/sec². The measuring equipment described is particularly suitable for measurements in the case of vehicles starting up or coming to a standstill in a period of a few seconds. Just before stopping, the pointer wavers somewhat because the frequency of the speed indication is too low, and the reading in this time interval is generally not utilisable. This blank interval can be reduced, however, by using an a.c. tacho-generator with a large number of poles, and fully utilising it at its maximum permissible speed. For calibration, the linearity of the tacho-generator e.m.f. can also be checked by comparing it with the rectified voltage in the measuring circuit.

GERMANY

Two New Instruments for Materials Testing

By K. RÜHL. (From *Die Technik*, Vol. 3, No. 7, July, 1948, pp. 301-304, 8 illustrations.)

VARIOUS types of static and dynamic strain gauges, vibrometers and accelerometers have been developed in Germany in recent years, for instance, the Pfender clock-gauge extensometer, the Thiele carbon resistance pick-up for compressive loads, the Kargl capacitive pick-up for low loads, the Maihak gauges based on the vibrating wire principle for tensile and compressive loads, special piezoelectric pick-ups, and the inductive strain gauges and accelerometers developed by the D.V.L., Dr. Lehr, and other investigators. Although a number of these pick-ups have already been described in American and British publications, descriptions are given in the following of two devices which have various interesting features.

1. INDUCTIVE ACCELEROMETER

The inductive type accelerometer developed by Wolff is shown in Fig. 1. It consists chiefly of two coils (*a*) fixed on a small cylinder mounted on the structural component under test, and a movable iron core (*b*) acting as an elastically suspended mass in the damping chamber (*c*) within the coils (*a*). Any accelerated or uniform motion of the iron core then generates an e.m.f. in the coils which is transmitted to an oscillograph or other indicating instrument.

In this connection, it is pointed out that further work will still be required to develop a pick-up with a very high frequency response suitable for impulsive loading, and not affecting (as in the case of quartz or carbon pick-ups) the quantity to be measured by closed-end reflection effects. Furthermore, owing to the shock wave propagation and distribution, the momentary stresses can be higher than those due to the corresponding statically applied forces. This problem is the subject of an as yet unpublished report by Dr. K. Fink, of the Iron Research Institute, Düsseldorf.

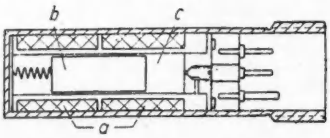

Fig. 1. **Inductive type accelerometer according to Wolff.** *a* coils, *b* iron core acting as an elastically suspended mass, *c* damping chamber.

2. CLOCK-GAUGE EXTENSOMETER

A simple clock-gauge type extensometer for the direct measurement of strains in framework components, etc., is the portable Pfender extensometer shown in Fig. 2. The change in the gauge length between two steel balls hammered or otherwise inserted into the test piece is obtained by re-adjusting the distance between two steel caps at the base of the extensometer. One of these caps is fixed on the extensometer body, while the other, by manual operation of a system of levers, can be rotated so that it will be displaced a certain distance away from the fixed cap and thus vary the gauge length. The lever motion is indicated by a clock gauge, which is calibrated to read either the actual variation from the original gauge length in centimetres or the relative variation, i.e., the strain in the test piece. The Pfender gauge has been used for the periodic checking of deformations occurring in bridges, etc., and can be inserted in not easily accessible members.

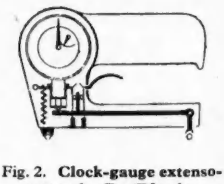

Fig. 2. **Clock-gauge extensometer by Dr. Pfender.**

DESIGN AND WORKSHOP

A Digest of articles of interest to Works Managers, Planning and Production Engineers, Superintendents, Jig and Tool Designers, Inspectors, Time Study Engineers, and all those concerned with the practical aspects of Engineering.

Milling High-Strength Alloys at Elevated Temperatures*

By A. O. SCHMIDT (Research Engineer), KEARNEY & TRECKER CORPORATION.

NEW and improved workpiece materials, having high strength properties, together with large volume of output demanded by mass production requirements often make metal cutting in general, and milling in particular, a difficult operation. There are two ways of milling high strength materials in order to obtain reasonable tool life—one involves reducing the cutting speed and feed and cooling the cutter, if possible; the other, increasing the cutting speed and feed but preheating the workpiece.

Generally, the hardness and the strength properties of a steel workpiece can be taken as an approximate index of machinability. Steel of 200 Bhn may be cut efficiently with a carbide cutter at a cutting speed of 400 fpm and a feed per tooth of 0·010 in. The same material heat treated to 300 Bhn would require a reduction in cutting speed to about 300 fpm and in feed to about 0·008 in. per tooth in order to obtain reasonable tool life. When the steel hardness is increased to 400 Bhn the machining problem becomes more difficult. Not only is it necessary to decrease the cutting speed to about 140 fpm and the feed to 0·004 in. per tooth, but the machine requirements become more stringent.

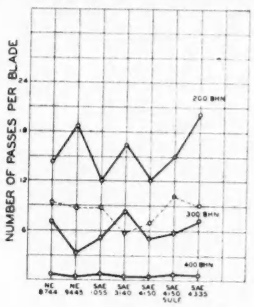

Fig. 1. **Average number of passes per carbide blade when milling steels of 200, 300 and 400 Bhn.**
Heavy lines indicate performance at a cutting speed of 428 fpm and a feed of 0·0115 inches per tooth. Dotted line represents steels of 400 Bhn milled at a cutting speed of 130 fpm and a feed of 0·0045 inches per tooth.

In Fig. 1 are plotted the average number of passes per carbide blade when machining different steel test bars 12 in. long, heat treated to various hardnesses. The milling cutter operated at 428 fpm cutting speed and a feed of 0·0115 in. per tooth. At this speed and feed the tool life of the carbide blades was very low when milling test bars of 400 Bhn, but by decreasing the cutting speed to 130 fpm and the feed to 0·0045 in. per tooth better results were obtained, and these are indicated by the dotted line.

The higher strength materials require more power per cubic inch of metal removed per minute, which also means that more heat is generated in the cutting operation. Sintered carbides and cast materials possess the property of keeping their hardness at elevated

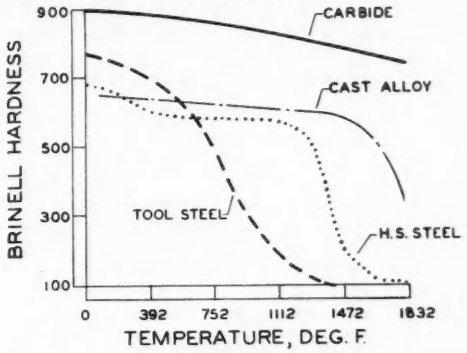

Fig. 2. **Effect of temperature upon the hardness of tool materials.**
Based on data by E. Amman. Hardness of cast material was estimated by performance in hot milling.

temperatures. (See Fig. 2). When cutting SAE 1050 of 200 Bhn at 400 fpm cutting speed, the temperature of the tool in the vicinity of the cutting edge will reach 1400° F after a few seconds, while the chips themselves will have a temperature of about 760° F. All this heat is generated with the workpiece near room temperature.

Since basically the machine tool is the inverse of a heat engine and is supplied with mechanical power which generates heat in the tool, chip, and workpiece, the less heat the machine tool is required to generate, the easier the metal cutting operation becomes. There are several ways to make machining easier, e.g., using materials which have inherently good machinability properties such as the light metals, additions to ferrous materials, heat treatment and, last, but not least, cutter design.

Another method of making steels and other materials more machinable is to cut them while they are at an elevated temperature, since hardness and strength properties are then reduced. This procedure is feasible in operations such as roughing out die forgings immediately after they have come from the forging die, or by preheating in a furnace in those cases in which the workpiece will be heat treated again.

When a number of old die blocks used in drop forging required resurfacing, the cutting speed for the carbide cutter had to be reduced to about 100 fpm and the feed to 0·0035 in. per tooth. Only thus was it possible to obtain a satisfactory tool life of 12 passes 24 inches long, 6 inches wide, and 0·250 in. deep with a 10 in. carbide cutter. At higher speeds and feeds the cutter would fail quite frequently after only one or two passes. The machine used in these tests was a Kearney & Trecker CSM bed type machine of high accuracy and rigidity.

*Specially contributed to THE ENGINEER'S DIGEST, this article is also scheduled for publication in the American journal *The Iron Age*.

The power requirements, as measured with a wattmeter, were very high. Another indication of the large amount of work required in this particular milling job was the high temperature of the chips, which would reach 1400° F, as determined by comparison of the chip colour with a heat-colour chart.

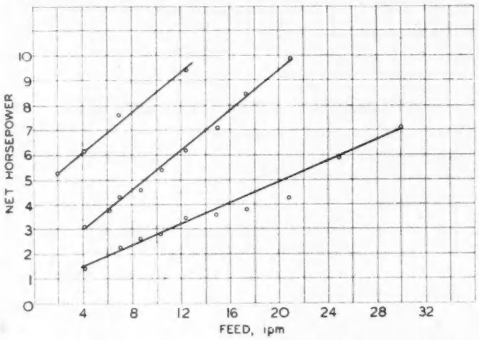

Fig. 3. **Horse-power required in milling a die block.**
Upper line indicates power to mill block in " as received " condition. Lower line shows power required when block was at a temperature of about 1500° F and centre line indicates power after block had cooled again to room temperature.

Fig. 4. **Milling cutter with solid sintered carbide teeth as used in tests for milling heated die blocks.**
The rounded chip space in the hardened cutter body is chromium plated facilitating unhampered chip flow.

Fig. 5. **Typical chips produced by the cutter shown in Fig. 4.**

In further machining tests, one of the die blocks was heated in a furnace to 1800° F and then mounted in an insulated fixture on the milling machine table. The milling of this heated die block started when the block was at about 1500° F, as determined by comparison with a heat-colour chart, and the temperature continued to drop slowly as the tests proceeded. The power required, measured with a wattmeter, is plotted as the lower line in Fig. 3. After the block had cooled to room temperature its hardness on the surface had dropped from an average of 400 Bhn to 350 Bhn, which also reduced the power needed as indicated by the middle line in Fig. 3. When the block was at room temperature in its original state, the power consumption was as shown by the upper line.

Due to the interrupted type of cut in milling, tool life is comparatively good when machining a heated workpiece, because the tooth has a chance to cool in the air between cuts. Changes in microstructure and accuracy in the workpiece are unavoidable under these conditions, but a difficult milling job can be made easier this way. Care should be taken to provide proper protection against the heat flow from the workpiece into the machine, either by insulating with layers of asbestos or by circulating a coolant in the table or fixture. Preheating the workpiece means that the cutter will not be required to generate as much heat as when the workpiece is at room temperature. Instead of heating the chip material from room temperature to about 1400° F by deformation, compression and friction attendant to normal cutting action, preheating the die block to the above temperature will require a much smaller percentage of additional heat to be generated during the chip formation.

In tool-life tests on cutters with carbide tips (see Fig. 4) the same number of passes was completed when milling heated workpieces at high feed rates as when milling the same workpieces at room temperature at only one-eighth the feed rate.

Although the workpieces used in these tests were heated in the furnace, preheating the workpiece on the milling machine itself is feasible and will permit increased feed rates for optimum production.

A large number of slab milling tests was conducted on a Kearney & Trecker 1808 CSM Simplex Machine. The cutter used was a Goddard & Goddard slab milling cutter 5 inches in diameter, 4 inches wide, and with 8 teeth tipped with Tantung. The workpiece was a flat plate of SAE 4340, 280 Bhn, with the surface preheated by an Airco flat surface torch, $3\frac{1}{2}$ inches wide and water cooled, see Figs. 6 and 7. The workpiece, chamfered at the sides, was held in the fixture by clamps acting on the chamfered surfaces. Pieces of asbestos board were used for insulation and for shielding the cutter and spindle nose. Before the table feed was engaged, the torch was lighted and the flame adjusted while the spindle was running. Feeds used in these tests were 50 and 60 ipm with a spindle speed of 180 rpm, equivalent to 236 fpm cutting speed. The heat supplied by the torch was sufficient to heat the surface of the workpiece to around 1200° F. This softened the material and thus cushioned the shock when the milling cutter tooth entered the work. The temperature of the work surface at the point of engagement with the cutter, several inches away from the torch, was about 1100° F. In addition, the power requirements were reduced as can be seen in the graphs, Figs. 8 and 9. There was no discoloration or temper colour on the milled surface which indicates that temperatures in the workpiece were below 350° F during the cut. A microscopic examination revealed no difference between the microstructure of a piece machined at room temperature and one machined with a preheated surface.

These numerous tests conducted with the workpieces heated on the milling machine with an oxy-acetylene torch indicate that, because of reduced power consumption, milling is thus possible at greatly increased feeds. Higher production would justify the additional expense incurred in preheating the workpiece.

When applying heat on the top surface of the workpiece, it was possible to control the heating in such a way that the impact blow between the cutter and work was reduced, the horse-power consumption lowered, and the rate of feed increased so that practically all of the heated material was cut out or removed in the form of chips. The workpiece and cutter were not too warm to be touched by hand immediately after the cut and no noticeable distortion of the workpiece could be detected.

As can be seen in Fig. 2, carbide remains comparatively hard at high temperatures and will, therefore, mill a heated workpiece under conditions which will

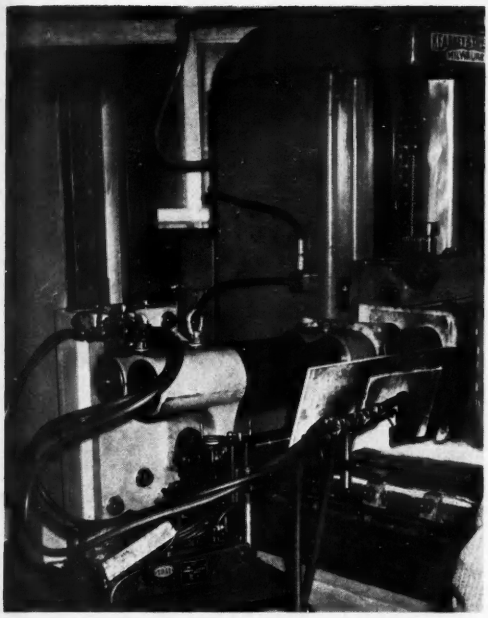

Fig. 6. **Bed type milling machine used for slab-milling tests.**

Airco Torch is preheating the workpiece. Cutter and spindle used are shielded by asbestos plate.

cause a H.S.S. cutter to fail almost immediately. Many tests were also run with cutters having teeth of cast material. Judging by the good performance of cast materials, their hardness when milling a heated steel piece will follow the line indicated in Fig. 2. Cast alloys prove to be satisfactory tool materials for use in hot milling.

Heating a steel workpiece will decrease the power

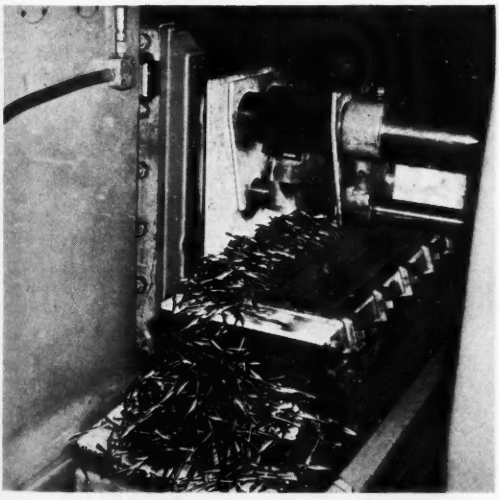

Fig. 7. **Milling the preheated workpiece of SAE 4340, 280 Bhn, with 8-tooth tantung slab mill.**

Cutting speed 236 fpm, depth of cut 0·200 in., width of cut 4 in., feed 60 ipm. 48 cu. in./min. of material were removed. Power required, 35 hp.

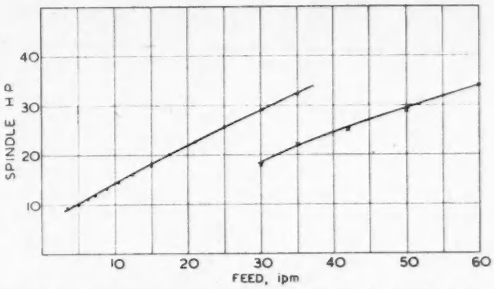

Fig. 8. **Spindle horse-power in relation to feed of milling machine shown in Figs. 6 and 7.**

Depth of cut 0·200 in. Lower line for preheated workpiece. Upper line for workpiece at room temperature.

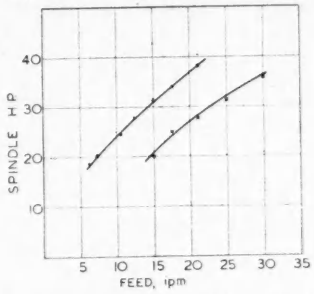

Fig. 9. **Spindle horse-power in relation to feed of milling machine shown in Figs. 6 and 7.**

Depth of cut 0·380 in. Lower line for preheated workpiece. Upper line for workpiece at room temperature.

requirements appreciably. Combined with intermittent cutting, an inherent characteristic of milling, which results in cutter-workpiece contact only part of the time, this practice also results in good tool life. Carbides and cast alloys are suitable tool materials having the desired properties at elevated temperatures, i.e., they remain harder than the material they are cutting and can thus function as metal cutting tools. Cutters must be designed for good chip flow and immediate ejection of the chip to avoid unnecessary accumulation of heat in the cutter. (See Figs. 4 and 5.)

Hot milling, or milling a workpiece the surface layer of which has been softened by the application of heat, quite obviously is not indicated as a desirable machining method when other, more established, procedures prove expedient. Application of this technique is at present limited to operations which cannot be performed practically or economically by the more common methods.

ACKNOWLEDGEMENT

Grateful acknowledgement is made to Mr. Russel Stuart of the Walker-Turner Division and Messrs. J. B. Armitage and J. R. Roubik of Kearney & Trecker Corporation for their encouragement and help in the tests reported herein.

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Barrel Burnishing of Light Alloy Products

By Ch. Etienne. (From *Revue de L'Aluminium, France*, Vol. 25, No. 150, December, 1948, pp. 377-382, and Vol. 26, Nos. 151 and 152, January and February, 1949, pp. 15-21, and 65-69, 45 illustrations.)

Mechanical lathe polishing is a skilled and costly finishing operation. For articles or components not requiring the highest standard of surface finish, barrel burnishing with steel balls is much cheaper. Parts with intricate details and shapes can easily be burnished by this method instead of being electro-plated or electro-polished. Barrel burnishing has been used to a large extent in the U.S.A. and Germany.

A. Surface Qualities

Articles to be burnished are placed in a wooden barrel to which steel balls and a lubricating liquid are added. The barrel is then rotated, the balls and articles thus being vigorously stirred and the balls impinge on the surfaces of the articles. The points of impact receive a brilliant finish; the higher the polish of the balls, the better the finish. After treatment, the surfaces consist of innumerable small dents giving a brilliant, though diffuse, reflection. The surface is also cold-worked and levelled out (Fig. 1),

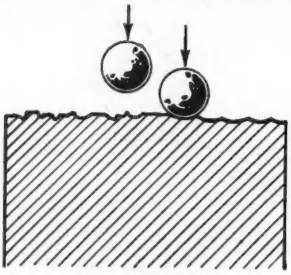


Fig. 1. **Working by ball hammering (burnishing).**

except for the very small "Brinell" impressions, the average diameters of which vary from 0·001 inch for hard alloys to 0·02 inch for pure aluminium under normal working conditions with balls of $^5/_{32}$ inch diameter. These impressions cover the whole surface of the part, except for inaccessible sharp corners and grooves (Fig. 2). Not all the light alloys are equally well suited for burnishing. Pure aluminium gives a highly brilliant finish, Duralumin or Al Mg Zn a good finish, and alloy AS 13 only a mediocre finish. The suitability of a material can be judged from a single impression under the Brinell press with a highly polished ball indenter. X-ray examination shows that, after burnishing, only the surface layer is slightly cold worked and the crystals are not so highly worked and distorted as after mechanical polishing. This difference explains the better resistance of the burnishing polish against wear, corrosion and handling. Burnished surfaces are not easily wetted by water.

B. The Burnishing Balls

The balls must possess great surface hardness and abrasion resistance, and a high and lasting polish. They are made of nickel-chromium steel, fully hardened, ground and lapped. Balls as used for second or third-grade ball bearings can be used. Balls with hardening cracks might break during burnishing and damage the articles. Before use, therefore, the balls are dropped onto a marble plate from a height of about 2½ feet, and cracked balls can thus be detected and eliminated. Cast or forged steel balls cannot be used, as they do not keep their polish long enough. Tungsten carbide balls as used in U.S.A. last practically indefinitely. The choice of diameter depends on the hardness of the alloy to be treated, smaller balls being chosen for harder alloys. Fragile articles are more

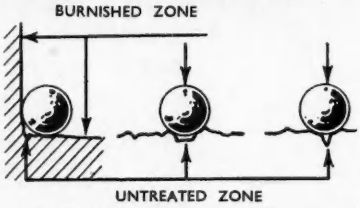
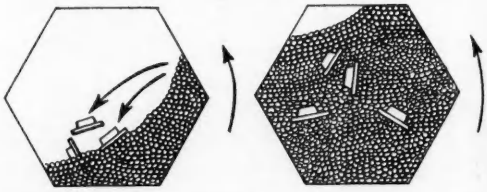


Fig. 2. **Influence of ball diameter on the accessibility of grooves and corners.**

easily deformed by small balls. The density of a bed of balls does not change with the diameter of the balls; therefore, for higher pressures, a larger height of bed is required, and short drums of large diameters are preferred for robust articles. Increased pressure increases reflecting quality and reduces time of treatment. The "viscosity" of a bed of balls, i.e., the resistance against displacement, increases with increasing diameter of the balls, but decreases appreciably if balls of different diameters are used, (Fig. 3). Reduced "viscosity" increases the rate of working and shortens the time of treatment. Light alloy parts of medium size are thus burnished in small barrels with balls of $^5/_{64}$ to $^5/_{32}$ inch diameter, and larger articles in larger barrels with $^5/_{32}$ to $^3/_{16}$ inch balls. The ratio of the volume of the bed of balls to the volume of treated parts varies from 1·5 to 2 for small items (up to 1 inch across) and from 5 to 7 for larger items (4 to 6 inches). If balls of different diameters are mixed, this ratio can be reduced, provided the parts remain covered by balls, and if possible, do not touch each other. The barrel is filled to about two-thirds its capacity, or less for certain fragile articles (Figs. 4 and 5).

Balls are supplied polished and greased. They must first be degreased by trichlorethylene treatment; their reflecting quality then being restored by stirring for 5 to 10 hours in a cold 10 per cent solution of sodium carbonate, followed by rinsing in cold water. A separate vessel should be used for this treatment so as to

Fig. 3. (Left) **Stable equilibrium of three balls of equal diameter.** (Right) **Addition of a smaller ball makes the equilibrium unstable and reduces resistance against displacement.**



Figs. 4 and 5. (Left) **Barrel not sufficiently filled. Pieces issue from the bed of balls, are thrown about and knock against each other.** (Right) **Barrel too full, movement is impeded and time of treatment increased.**

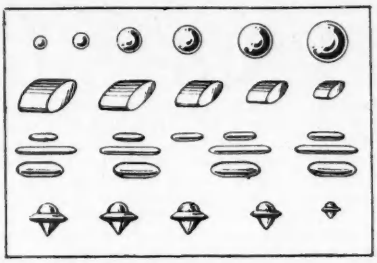


Fig. 6. **Various shapes of burnishing agents used in practice.**

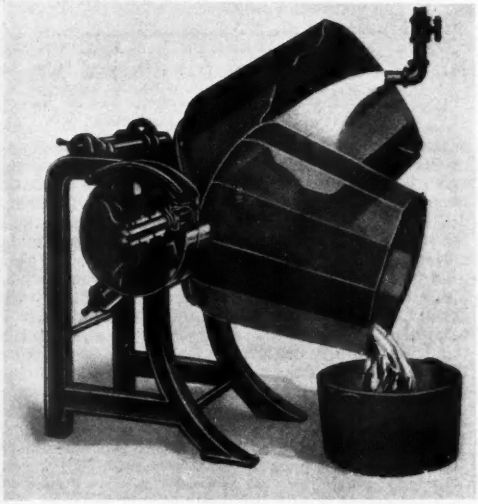


Fig. 7. **Burnishing barrel with inclined axis shown in working and unloading positions.**

avoid contamination of the burnishing liquid. The balls can be stored in wooden, earthenware, or enamelled containers under a 5 per cent soap solution in order to prevent oxidation or corrosion. The solution should be stirred every 2 days to stop gelatinization. During a brief work interval, the balls are left in the barrel and entirely covered with burnishing liquid. Before resumption of work, the barrel is then rotated for about 10 min. Reflecting quality can be restored by the sodium carbonate treatment whenever required. Various forms of burnishing agents are used and often mixed (Fig. 6). Some of these forms are especially suited for burnishing complicated and delicate shapes.

C. The Barrel

Barrels are usually made from hard wood, maple and beech being the best; plane and poplar are also suitable but less resistant. Resinous woods (pine, fir) and tannin-containing woods (oak, chestnut) affect the liquid and tarnish the articles. Steel barrels with an internal timber or neoprene lining are used for very large drums. Barrels of 1 foot diameter are usually of hexagonal shape, larger barrels of 2 feet diameter are octagonal. (Figs. 7 and 8). The number of edges determines the number of stirring movements of the balls, but these movements are more vigorous, the sharper the angles. The ratio of length to diameter is determined by the required height of the bed of balls and is usually small, so that a great height (i.e., increased pressure and quicker treatment) is obtained with a smaller amount of balls. It is about 0·4 for the treatment of hard and robust parts (steel, bronze) in 2-ft. barrels. Light alloy articles should be treated in 1 ft. barrels having a ratio of 2 to 2·5, so that many articles are treated simultaneously under a smaller ball pressure. Rotation speeds depend on the ball diameters and the shape and size of the barrel. An average speed of 3 fps measured on the outside of the barrel is recommended. Too high a speed leads to rebounding of the balls, producing a characteristic noise, and too low a speed leads to excessive duration of treatment. Special barrels are often used for small and delicate articles, which are either attached to the drum walls or strung on wires. This design protects the parts against deformation and facilitates sorting after treatment.

D. The Burnishing Liquid

The liquid

(a) should lubricate balls and articles;

(b) should not form adherent films, which would have to be washed off in a special operation and would attract impurities. Liquids based on mineral or vegetable oils should not be used, emulsions being preferable;

(c) should not chemically attack or tarnish the articles or balls, i.e., should not be strongly alkaline and should not contain a large amount of mineral salts;

(d) should not attack the wood;

(e) should be non-toxic. If a sulphur base is used, the operators must wear rubber gloves;

(f) should last for at least three charges of the most unfavourable metals (Al-Si alloys);

(g) should be easy to procure and store. It must not pick up dust when left to stand and should neither form deposits nor change in concentration by evaporation. Emulsions must be perfectly stabilised;

(h) should be cheap. An aqueous solution is, therefore, preferred.

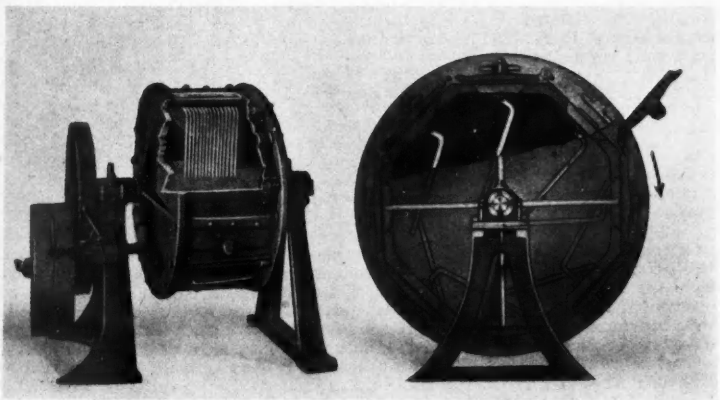


Fig. 8. **Cylindrical barrel with octagonal inside shape, transverse axis of rotation, and mobile rakes to stir the bed of balls.**

5 to 10 per cent aqueous soap solutions based on non-resinous vegetable oils are normally used. Copra and palm oil soaps give the best lather; olein and saponin are suitable. Soaps are supplied in tablet, powder or granular form, dissolved in hot water and left to cool. Alkalinity is controlled by additions of oxalic or tartaric acid where necessary. Soap quality, however, is often irregular. Other liquids, based on essence of Panama wood or mineral salts, have been used with some measure of success. The best composition, as recommended by the author, is: 35 pints distilled water, $\frac{7}{8}$ pint benzine, $\frac{7}{8}$ pint ammonia, $\frac{7}{8}$ oz. polishing rock soap. This emulsion gives consistently good brilliant surfaces after only 30 minutes treatment, is stable and dissolves grease. It contains only 0·1 per cent soap and is, therefore, cheap and insensitive to changes in alkalinity. Its ammoniacal pH of 12 is neutral for aluminium and falls to 10 after heavy use of the bath. Storing is easy, no deposit is formed and any aluminium dust can be filtered out. Ammonia may, however, evaporate during prolonged period of storage. As the liquid does not form films, neutralised articles dry rapidly, and no stains are left. Normal hard water precipitates the soap and prevents formation of a persistent lather; distilled water should, therefore, be used. Commercial benzine and ammonia may be used; rock soaps of low pH are best.

The number of charges that can be treated without change of bath depends on :—

(a) The cleanliness of the articles.

(b) The material. Pure Al can be treated without difficulty; Al-Mg and Al-Cu alloys must be perfectly neutralised after cleaning; Al-Si alloys receive special treatment.

(c) The method of manufacture. The rougher the surface, the more difficult is the removal of oxide or Si-skins after degreasing. Intricate castings are the most difficult to clean and make heavy demands on the burnishing bath; die-castings are better, and forged or machined articles very much superior.

(d) The volume of the charge. In larger barrels, the volume of liquid decreases relative to the volume of articles in the charge and the liquid is used up more quickly. One bath was found to last for 10 charges in a barrel of $8\frac{1}{2}$ inches diameter and 20 inches length, but only for 5 charges in a 16 × 24 inch barrel.

(e) The composition of the bath. 10 per cent soap solutions have a shorter life than the recommended emulsion which, in a 9 × 20 inch barrel, can be used for 15 charges of pure Al-articles, 8 charges of Al-Cu, and 3 charges of Al-Si articles, provided the articles are properly cleaned.

The liquid should cover the entire charge by a depth of about 2 inches. With a 10 per cent soap solution, however, good results are obtained even if the liquid does not completely cover the charge. Soap can thus be saved.

E. Preparation of the Products

(a) *Buffing*

Cast parts should first receive an abrasion treatment so that a good surface may be obtained. Since burnishing does not eliminate abrasion scratches, the articles must then be buffed; the buffing should be carried out for a longer period, the rougher the cast surface and the higher the final polish required. Very carefully die-cast articles can, with some degree of success, be burnished without abrasion treatment. Buffing of sand-cast articles should be commenced with a fairly rough-grained wheel (No. 120) and finished with a fine-grained wheel (No. 140 or 160). Chill or die-cast articles can be treated with one fine-grained wheel. Soft alloys should be finished off with a No. 140 wheel and hard alloys with a No. 160 wheel. Buffing discs should consist of Si-carbide with a hard binder; this should be harder, the softer the alloy. Other abrasive materials wear too quickly.

Plastically worked or machined parts need no abrasion treatment unless the working has left noticeable scratches or the material is very coarse-grained. A light treatment with a No. 160 wheel is then sufficient. Small intricate parts cannot be buffed, but may be treated in bulk by "tumbling" with corundum abrasive. Care must be taken to avoid non-uniform wear of the parts. The abrasion treatment is very expensive and, if possible, should be avoided by careful design and manufacture.

(*To be continued*)

Novel Facing Tool for Radial Drills

(From *Werkstatt und Betrieb, Germany*, Vol. 82, No. 1, January, 1949, p. 13, 1 illustration.)

With the tool shown in Fig. 1, a downward movement of the drilling machine spindle causes a corresponding upward movement of the facing cutter. The toolholder *a* fits into a quick-change chuck of the radial drill and carries the sliding bush *b* inside which slides the shaft *c* with an extension shaft *d* carrying the facing cutter *s*. Pin *i* fitted in bush *b* carries a small spur gear *r* which meshes with racks *Z*1 in holder *a* and *Z*2 fastened to shaft *c*. The compression spring *f* first presses bush *b* downwards until the movement is stopped by set-screw *h*. The tool is now inserted in the drill jig *v* until the locating bush *e* comes up against the guide bush of the jig. The facing cutter is then fixed to extension shaft *d* by means of a bayonet grip.

a = toolholder
b = sliding bush
c = shaft
d = extension shaft
e = locating bush
f = compression spring
h = set-screw
i = pin
m = nut limiting cutter lift
r = spur gear
s = facing cutter
*Z*1 and *Z*2 = racks
v = drill jig
W = work piece.

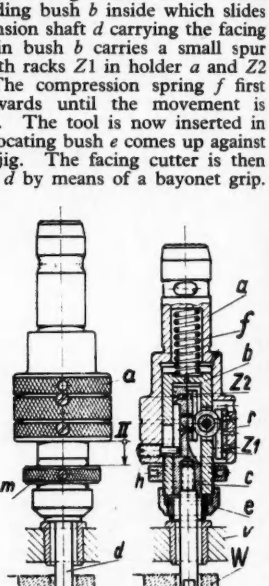


Fig. 1

If the toolholder *a* is now pressed down again, the movement is transmitted to spur gear *r* and reversed by rack *Z*2 so that the facing cutter is raised against the bottom face to be machined. Nut *m* can be adjusted to limit the cutter lift. The tool described here allows of drilling and facing a work piece without re-clamping and avoids an upward movement of the spindle during facing, thus making it impossible for the toolholder to slip out of the drill chuck.

CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns, are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

BOILERS

The Calculation of Boiler Feed Water Heating

By S. S. GHEORGHIU. (From *Buletinul Institului Nationzl de Cercetari Tehnologice, Rumania*, Vol. 3, No. 3-4, 1948, pp. 376-399, 9 illustrations.)

THE heating of feed water by bled steam permits an important increase in the efficiency of boilers, without necessitating costly installations or greatly complicating the operation of boilers. The gain in thermal efficiency depends on the characteristics of the heating process, that is, the initial steam conditions, the type of heat exchanger employed, the number of heating stages and the extraction points at the turbine. It is important to arrange the whole installation in such a way that the decreased use of waste heat in economisers and elsewhere does not result in losses which have to be offset against the improvements possible.

In this study, the effects of initial conditions, which are well known, are not given any further consideration. It is endeavoured, on the other hand, to give a method of calculation, suitable for practical use, of the most beneficial choice of conditions for the bled steam, the total heat gain, the gain in thermal efficiency and the relative heating efficiency (compared to the ideal heater with an infinite number of stages) in terms of the number of stages adopted and the initial steam conditions. To this end, use is made of existing theoretical data, and three steps of approximation are carried out, each with fairly great precision. The cumulative error is hoped to be negligible in practice. The final method involves detailed calculations (by means of Mollier diagrams) of only one stage and estimations of the ideal efficiency with an infinite number of feed heating stages. The improvements to be expected from a finite number of stages can then be found with comparative ease by graphic methods, which are described in detail.

METALLURGICAL ENGINEERING

Progress in the Study of Surface Oxidation at Higher Temperatures

By E. GULBRANSEN. (From *Revue de Metallurgie, France*, Vol. 45, Nos. 7 and 8, July and August, 1948, pp. 181-204, and 287-300, 39 illustrations.)

FIVE typical reactions must be investigated in a study of oxidation : (1) de-gassing of the metal, which may influence the speed of oxidation. By heating in vacuo after de-gassing, the influence of this reaction can be elimited. (2) Oxidation proper. (3) Hydrogen reduction. (4) De-carburisation. Reaction (3) or (4) allows of eliminating the oxide layer and of studying the metal underneath the oxide skin. (5) Volatilisation of the oxide layer, an important factor in the oxidation of Mo or W. The speed of oxidation can be studied by vacuum micro-balance, whilst electronic diffraction permits research into the atomic lattice and the primary (crystallite) structure of the oxide layer. The electron-microscope is used for a study of the primary and secondary (grain) structure and spectrographic analysis serves to determine atomic constituents of the oxide layer. Apparatus, testing methods, test pieces and results obtained are described in great detail.

The following picture of the oxidation process emerges. With the metals under test (Fe, Cr, Co, Ni, Fe-Cr 13, 18-8 stainless steel, Mo, W, Al, Mg) the oxide layers reach a thickness of 200 to 2000 Angström, according to time and temperature of oxidation. The bottom layer shows an irregular crystal form and mosaic structure, whilst in a second layer, the usual crystal lattices of the particular oxide are generally formed. The crystal size also increases with time and temperature of exposure. Dimensions and crystal shapes depend on the speeds of seeding and crystal growth. If these are high enough, a two-dimensional mosaic structure forms the surface and the crystals are forced to grow in a direction normal to the surface, or by coalescence. The speeds of seeding and crystal growth are exponential functions of the free activation energy, the particular oxide and the exposed orientation, and the temperature. Number and size of the oxide crystals thus depend on temperature ; shape and other characteristics are determined by the speed of formation and growth and by the crystalline structure. Oxide skins are usually not uniform in thickness ; they consist of several layers. Seeding thus starts not only from contact with the initial oxygen film, but also from contact with other oxides. The outer layers, therefore, show a more irregular crystal grouping and unequal thicknesses. Local impurities and small free spaces in the initial film contribute to this phenomenon. After the first skin with irregular crystals has been formed, a second and third skin often form on top of the first one. These grow under less restricted conditions and form more regular crystals.

If the volume of the unit crystal of the oxide is greater than that of the metal, the oxide usually forms a protective layer against further oxidation. Assuming that the oxide is allowed to form uniformly in all directions, one would expect compression forces in this type of oxide skin. However, if the unit volume of the oxide crystal is smaller than that of the metal, tensile stresses can be expected to lead to porosity or hairline cracks in the oxide skin. Fig. 1 shows the mechanism of formation of the first oxide skin. The first reaction between metal and oxygen is the formation of a mono-molecular, uniform, film of oxygen. A thin layer of oxide then develops slowly or quickly, according to temperature. At a certain point, seeding sets in and then the crystals start to grow. Growth along the surface is limited by the simultaneous formation of many other crystals.

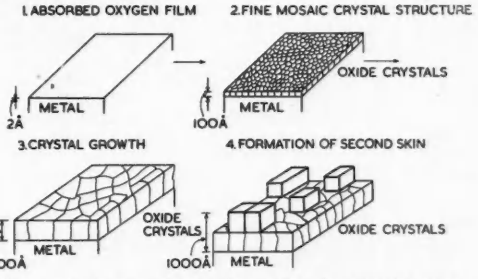


Fig. 1. Mechanism of oxidation. Secondary (grain) structure of oxide as indicated by the electron microscope. (× 50,000).

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In the current theory of oxidation, continuous and uniformly thick oxide layers are assumed to exist. It has been found, however, that oxide films consist of small crystals of 200 to 2000 Å extension, and crystal form and boundaries must play an important role in oxidation phenomena. Perhaps a recent improvement in the resolution of the electron microscope, down to 10 Å, will enable a study to be made of the influence of the crystal boundaries. With the present resolution of 50 Å, the crystal dimensions are at the limit of the resolving power of the electron microscope.

SCIENTIFIC INSTRUMENTS

Presentation and Measurement of Surface Roughness with the Aid of Electron-Microscopes

By R. SEELIGER. (From *Metalloberfläche, Germany*, Vol. 3, No. 1, January, 1949, pp. 9-14, 12 illustrations.)

To satisfy the strictest requirements regarding surface smoothness and accuracy, a special electron-microscope technique of presenting and measuring surfaces and relief depths has been developed. Electron-microscopic presentation of surfaces can be effected by means of very thin transparent detachable layers (oxides, lacquers, quartz or metal evaporation) deposited on the surface to be measured, or by means of the matrix technique, by which an imprint of the surface to be measured is made in a softer material and the detachable layer is applied to this imprint, or, alternatively, by a combination of the oxide and lacquer techniques. For very accurate measurements with the optical microscope, a straight and narrow beam of light is projected onto the surface and observed through a microscope, the axis of which forms a very small angle with the surface to be measured. The ridges of the line of light then represent the surface roughness and can be measured. A coarse estimation of roughnesses by means of electron-microscopic pictures is possible with the lacquer method or the evaporation method. With the former method, it is doubtful whether the lacquer layer really fills in all the cavities of the surface; the latter method affords an approximate measurement of the roughness from the lengths of shadows and the angle of spattering.

The high definition of depth of electron-microscopic pictures permits production of stereoscopic pictures, from which the distance of corresponding points can be measured and the depth measurements plotted in a graph. Another new method is an application of the "shadow-casting" method to the electron-microscope. An oxide layer is spattered with iron, at an angle of 45 degrees, from a white-hot iron wire. A glass thread is arranged exactly parallel to the iron wire and at as small a distance as possible from the oxide impression. The iron is spattered onto the oxide film and renders it opaque to electrons, except where the glass thread shadows the impression. The border-line between opaque and transparent regions of the oxide impression then shows up the ridges and depths of the surface and represents an oblique section through the surface, from which direct measurements can be taken.

STRENGTH OF MATERIALS

On the Fatigue Limits of Notched Specimens

By TADASI ISIBASI. (From *Memoirs of the Faculty of Engineering, Kyushu University, Japan*, Vol. 11, No. 1, 1948, 31 pages, 17 illustrations.)

IN notched specimens under repeated load there are two kinds of fatigue fractures, the first of which is preceded by an appearance of a small yielded zone or zones under stress equal to the fatigue limit, while in the case of the second, the peak stress at the root of the notch is less than the yield point of the material itself and no yielded zone appears until the specimen breaks. In the former case the notch-factor β turns out to be very small compared with its form-factor α.

To explain this and other experimental facts concerning the relations between α and β it is assumed that yielding or fracture of a point in a material is determined by the stress condition of a second point at a distance ϵ_0 (in the case of yielding) or ϵ_0' (in the case of fracture without any appearance of yielded zone) from the point under consideration. With this assumption, the following facts were made clear.

(*a*) The notch-factor is always smaller than the form factor of the notch.

(*b*) In notched specimens of a similar character, the reciprocals of the fatigue limits and the curvatures of the root of the notch are in parabolic relation.

(*c*) Appearance or non-appearance of a small yielded zone at the roo. of the notch, when a notched specimen is stressed to its fatigue limit, will be determined by the stress gradient at the root of the notch and the difference between the lengths ϵ_0 and ϵ_0', as well as the difference between magnitudes of the yield point and the fatigue limit of a plain specimen.

(*d*) In sharp-notched specimens, in which a yielded zone appears at the stress equal to the fatigue limit, the notch factor is extremely small, while in unsharp-notched specimens, in which no yielded zone appears at the stress equal to fatigue limit, the notch-factor is slightly smaller than the form factor.

(*e*) The characteristics of ϵ_0 and ϵ_0' are discussed and it is found that they are both linear functions of the notch radius for notched specimens of similar character.

(*f*) The lengths ϵ_0 and ϵ_0' are of the order of one hundredth or one tenth of one millimetre. Some connection may be found between these quantities and grain sizes of the material, but it is unknown as yet.

WELDING

Welding Rods for Acetylene Butt-Welding of Aluminium Alloys

By F. ERDMANN-JESNITZER. (From *Werkstatt und Betrieb, Germany*, Vol. 81, No. 9, September, 1948, pp. 248-251, 5 illustrations.)

IN the construction of light alloy containers and pipe lines, components made in different light alloys (e.g., tubes and castings) have to be welded together. A welding wire of 5 per cent Silicon aluminium alloy is often used because of its low melting point and good adhesion qualities. However, the strength of welds is not always satisfactory and points of leakage cannot always be repaired, since the weld remains porous.

In order to find a better welding material, seven different Al alloy materials were welded with different welding wires, and it was found that the following welding material gave better strength results without detriment to the metal structure: 3·6 per cent Mg, 0·6 per cent Si, 0·45 per cent Mn, remainder Al. Impurities: Fe below 0·5 per cent, Cu below 0·1 per cent, Zn below 0·2 per cent. It is recommended that quite general use should be made of this material for acetylene butt-welding. Soft Duralumin sheet (Al Cu Mg) of $\frac{1}{16}$ inch thickness is very sensitive to the quality of the weld material. Al Si 5 wire can be used for butt and edge-welding of Duralumin sheet; the 3·6 per cent Mg-alloy recommended can only be used for butt-welding, as is the case with the alloy Al Mg Mn. Pure Al wires without Si, and Al Mg 5 wires, are disadvantageous. A cross-weld test piece of Duralumin sheet showed an appreciable tendency towards tearing of the weld.

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BRITISH INDUSTRIES FAIR, 1949

A FURTHER REVIEW OF EXHIBITS

In our previous issue we have only been able to cover a limited number of exhibits to indicate the impressive results of Britain's productive efforts over the past 12 months. In the following, we review some further outstanding exhibits for the benefit of those who have been unable to attend the Fair. It will be appreciated that again owing to limitation of space, we have not included as many descriptions as we would have wished.

A.C.E. Machinery Limited, London, were exhibiting at Birmingham, and this year they concentrated on new productions of particular interest for works and factory installations. These new designs include push button operated platform and hopper hoists and two of the main exhibits were working equipments of this type for vertical hoisting of bulk material, drums, cases, sacks, etc. The range covers 2-10 cwt. capacity and the design is a distinct departure from the range of "ACE" standard design of Skip and Platform Hoists for larger capacities up to 2½ tons. To summarise, the exhibits included :—A push button, electrically operated Swivelling Hopper Hoist of the type supplied to the Food Industry; a push button, electrically operated, Dual Carriage Hoist for handling kegs, drums, cases, sacks, etc.; a 5 cwt. capacity General Purpose Portable Winch Unit of compact and lightweight design for works, farm or building duty; a Vibratory Screen representative of an extensive range. All exhibits were working equipments for demonstration purposes.

Aiton & Co., Ltd., Derby, were showing, at Birmingham, examples of plain, corrugated and creased steam pipes for use in Power Stations, ships and locomotives under the highest pressures and temperatures. This firm also showed a 27 in. bore by 2 in. thick by 15 ft. 3 in. long forged steel seamless steam receiver for one of the latest British Electricity Authority Stations, which showed examples of branch welding and reinforcing. The branches were fitted with "Corwel" joints. One of this firm's most interesting exhibits was a model triple effect P. & B. Evaporator Distiller, 1/10th of full size. These plants are designed to produce boiler feed make-up and for all purposes where distilled water of the highest purity is required. Other exhibits included Aiton forged steel steam traps and examples of bellows expansion joints which are used for taking up expansion in low-pressure pipe lines.

Birlec Ltd. were displaying at Birmingham a particularly interesting exhibit, namely, a fully automatic rotary-drum type furnace, designed for carburising small parts in quantity. This unit is fully mechanised to reduce labour to an absolute minimum. Charging, tumbling of the charge to ensure complete uniformity of treatment, and discharge direct into the quench tank are all made automatic, remote-controlled operations, so that one attendant can supervise the working of a battery of such furnaces without difficulty and with complete regularity of output and uniformity of results. This type of furnace can be used also for carburising in compound, if desired, without involving packing of the work in boxes.

Another type of specialised Birlec heat treatment was exemplified by an automatic high frequency induction heating unit for surface hardening automobile gear-box shafts. This unit was demonstrated on the Stand in actual production and showed the high rate of output obtainable from a compact machine and the simple, largely automatic operation of the process.

Other Birlec equipment, ranging from large steel-melting furnaces to bright annealing and other copper brazing equipment, was represented by models and photographic displays and by samples of typical furnace products.

Alfred Bullows & Sons Ltd., Walsall, Staffs., were showing, at Birmingham, two main exhibits; they were an entirely new design of Waterwash Paint Spray Booth and an Air Operated Agitator or Mixer, and were both working exhibits. The Spray Booth incorporates features reducing maintenance to a small fraction of that required on earlier types. The Booth exhibited was internally illuminated with glass front panels, so that its internal features could be examined while it was at work. An automatic spray painting unit was installed in the Booth, so that its effectiveness in removing paint from the exhaust air stream could be demonstrated. The Air Operated Agitator was mounted on a transparent vessel containing solids in suspension so that the precise action could be seen and understood. A range of multi-cylinder air cooled air compressors was also exhibited, a new development in this range being the three stage high pressure machine, developed primarily for operation of air blast E.H.T. switchgear. In addition, an extensive range of spray painting equipment was exhibited.

The Copper Development Association's Stand was restricted in scope to the display of a few of the important uses of copper and copper-bearing materials in such recognised fields as electrical and mechanical engineering, chemical engineering and building technique, and in less spectacular but equally interesting manufactures, such as boots and brushes, clock parts and spectacle frames.

A mechanical engineering panel demonstrated the suitability of copper, brass and bronze for burners and nozzles. A notable example shown was a low air pressure oil burner, such as is used for industrial furnaces and for steam raising in ships, power stations and factories. Other items displayed were welding and cutting blowpipes, a copper ring-burner for pressure welding, besides a copper tuyere for blast furnaces, etc. Two metallurgical panels demonstrated the variety of intricately shaped articles now produced by modern techniques in the cold pressing and the hot forging and stamping of brass.

The electrical industries are the greatest users of copper and copper alloys, and the appropriate display panel related to one of the innumerable applications, namely, copper in current-collecting gear. The panel devoted to wirework was of interest to the layman as well as the expert, as it illustrated little known facts about the use of copper wire in articles of everyday use.

Deloro Stellite Limited, Birmingham, were exhibiting, at Birmingham, a range of precision cast dies for the hot extrusion of copper and zinc base alloys. A variety of other precision cast components such as cams, dies, high pressure air valves, etc., was also shown. A feature of the display was that all this Company's products, i.e., STELLITE cutting metal, hardfacing rod and castings, are now produced in England as well as in Canada. A range of cutting tools for turning, milling, planing and woodcutting, was displayed. A number of STELLITE hard-faced components was also on view, including such items as steam and chemical valves, punches and dies, and various other components. Demonstrations of hard-facing with STELLITE rod were given throughout the

exhibition and visitors were cordially invited to bring their own small components for processing on the stand.

The English Steel Corporation Ltd. displayed at their Stand a hollow-forged high-pressure boiler drum. Complete with nozzles, tube holes and manhole covers, this drum is 26 ft. long, 3 ft. bore and weighs over 13 tons. Although limitations of space permitted the showing of only a small drum, it was a useful example of the hollow-forged type of boiler drum with bores up to 5 ft. diameter and weighing up to 65 tons. Another striking exhibit was a series of aeroplane and lorry crankshafts, finish-machined by E.S.C. from their own drop forgings. Exhibited for the first time was a range of Rock Bits for the deep drilling of boreholes for oil. Forged Steel kellies, drill collars and substitutes were also shown. These are made by E.S.C. under licence from English Drilling Equipment Company of Bilbao House, 36, New Broad Street, E.C.2, and as in the past they have been almost entirely a monopoly of U.S.A., their production in this country is a great dollar-saver. Briefly, the rock bit, seen here in Fig. 24, consists of a steel body in which rotate several sharp-toothed cutters, the rotary motion of the cutters being obtained from the pressure of the entire revolving bit on the rock. Most of the parts are drop-forged from high quality steel. The cutters rotate on ball and roller bearings, and to resist wear tungsten carbide is welded to the cutter teeth. The boreholes in which the bits work are sometimes more than 16,000 ft. deep, and to undertake such costly work it will be appreciated that the performance of the rock bit is more important than its cost. The British-made bits have been found comparable in every way with their American counterparts. Springs for road vehicles were also shown, although space did not permit the exhibition of the entire range of E.S.C. coil and laminated springs which are produced in a very modern plant. To round off the Stand visitors saw a display of Engineers' Small Tools covering almost the entire range of this type of product.

Fig. 24

Ferranti Ltd. were showing at Birmingham an interesting range of their products. Of particular interest were the products of the Ferranti Transformer Department. A full scale model of a large High Voltage Power Transformer was erected on the Stand, in connection with which there was a working demonstration of Rotary Type On-Load Tap Change Gear complete with control and ancillary equipment. Another interesting exhibit was a 5000 kVA, 11 kV Moving Coil Voltage Regulator. Of outstanding interest, however, was one of the largest bushings used in power transmission. This bushing is 16 ft. in height and weighs approximately 800 lbs. and portrays the trend in high voltage transmission.

A full range of Ferranti A.C. and D.C. High Service and Switchboard Meters, Prepayment Meters, Maximum Demand Meters, Two-rate Meters and Meter Testing Equipment was also shown. Of particular interest was the latest type of Ferranti Polyphase Single Disc Meter, which was shown side by side with a Polyphase Meter which was first put into service in 1906.

In addition, a complete range of Ferranti Instruments was on view, including the new 6 in. Square Dial Instrument, type H Self-Starting Motor with Gear Reduction Unit, and the Ferranti Volt-Ammeter, which had not been shown before in this country. The new range of Ferranti Miniaturised Hermetically Sealed Transformers was also exhibited. Other Ferranti Instruments included Hermetically Sealed Instruments for tropical use, Maximum Demand Alarms, Pocket Light Testers, Pilots' Crossed Pointer Indicator, and Ring Type Current Transformers. There were interesting working demonstrations of the Ferranti Phasing-Out Equipment, Summation Metering Equipment, Clip-On Ammeters, and an F.M. Relay and Current Transformer demonstration of Earth Leakage Protection.

Among the more interesting new developments of the Ferranti Electronics Department was shown a new 12 in. Cathode Ray Tube in heat resisting glass having an almost flat face due to a new glass manufacturing technique.

The Ferranti Servodyne is an electronic equipment, intended for use as a controlling medium for processes which require positive and precise control. The model of the Servodyne demonstrated was intended to illustrate the sensitivity and high rate of follow when observing, photo-electrically, different coloured lines on opaque or transparent backgrounds or the edges of materials. In this connection, the equipment has been successfully applied as a web guider on slitting machines in the paper converting industry, and a cloth guider on pin stenters in the textile finishing industry.

In view of the increased demand for a wide range of flameproof equipment for use in explosive surroundings, **The General Electric Company's** stand at the Castle Bromwich Section featured a selection of heavy-duty flameproof machinery suitable for oilfields, refineries, tankers, chemical plants, collieries and similar locations. These exhibits included Flameproof Oil Immersed Reversing Contactors and a Portable Mining Sub-station.

Among the exhibits was a type F.R. flameproof reverser, suitable for controlling the stator circuit of alternating current slipring motors up to 500 h.p. on colliery haulage and similar services requiring frequent starting, stopping and reversing. A flameproof transformer with H.T. fuses is fitted to the side of the case to provide 110 volts supply for the operating coils, the coils being energised from a type F.M.C. master controller, so that rapid opening and closing of the H.T. oil immersed reverser contacts is assured irrespective of the speed of operation of the lever controlling the rotor circuit.

Other exhibits included two oil circuit breaker units, coupled together to form a two-unit switchboard. These switchgears are available for either High Tension or Low Tension circuits and have been developed to prevent the exposure of live metal to atmosphere.

The near half of the Stand was devoted to a display of some typical examples from the vast range of decorative and industrial fluorescent fittings now available.

The main structure of the Stand occupied by **High Duty Alloys Ltd.** this year was built entirely of extruded "Hiduminium" section and incorporated a gate in wrought strip. Items displayed were limited to four, viz., Double-sided Impeller for Rolls-Royce "Nene," United Dairies Main Base for Filler, Crankcase for Bristol "Centaurus," and Dodgem Base for Supercar Co. (Coventry) Ltd.

These exhibits were supported by a series of photographs illustrating various production activities such as casting, forging, stamping, pressing and extruding, and the items produced by these methods in the Company's three factories.

E. Green & Son Ltd., Wakefield, were exhibiting their Economiser with cast-iron gilled tubes (Premier Diamond Type 25) representing the latest development in modern boiler-house practice. The Type 25 Premier Diamond Economiser, shown in Fig. 25, can be made suitable for working pressures up to 650

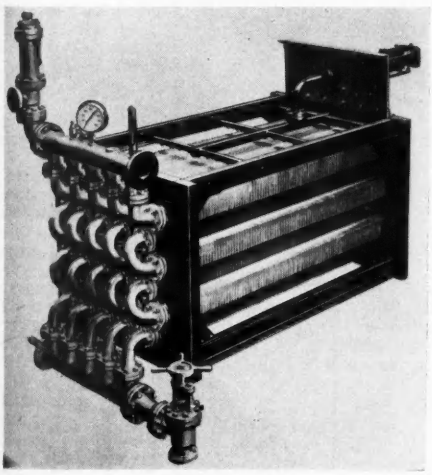

Fig. 25

lb. per sq. in., and has found wide application in all branches of industry, as well as in the smaller electricity generating plants. For working pressures beyond the limit of cast-iron Economisers, the diamond or streamline tube form having steel inner tubers is also available. Also on show was Green's Premier Diamond Type 12 Economiser, with cast-iron gilled sleeves shrunk on steel tubes arranged in shallow tiers, which combines the advantages of extended and compact heating surface, controlled gas flow, and complete facilities for inspection and cleaning with a steel tube construction suitable for all working pressures. Green's Cast-Iron Tube Airheaters, which provide clean hot air for heating, drying or combustion, were also included among the exhibits. They may be arranged to use waste heat from flue gases or can be direct-fired with a self-contained furnace. Green's Cast-Iron Triple Tube Airheater was also shown.

Lockers (Engineers) Limited were exhibiting, at Birmingham, machines from their range of "Locker-Traylor" Electric Vibrating Feeders, Screens and Conveyors. In addition, two types of mechanical vibrating screens were shown—the "Locker-Tital" for heavy duty work and also Locker "Rotex" Screeners for making fine particle separations according to size, on from one to five screen surfaces, in a single operation.

A further exhibit was the "Locker-Waytrol" Constant Weight Feeder for feeding granular material at any predetermined rate with remote control indication and totalising of the load. Also to be seen on this Stand was another range of products associated with this Company—the "Air-Maze" Air Filters, as well as Woven Wire Gauze from 2 to 200 mesh, woven wire screens, perforated metals with many shapes of perforations in many different materials, and wedge wire sieves. The exhibits included the Locker "Tufite" High Tensile Wire Screens, strong Wire Machinery Guards, and all kinds of crimped and welded wirework.

Valves covering the domestic field and those suitable for communications and industry have been exhibited by **Mullard Electronic Products Ltd.** at Olympia. Outstanding among the exhibits in the domestic radio class were the new series of valves on the B8A type base. Exhibits for communications and industry included, in addition to several normal comprehensive ranges, many interesting valves and electron tubes which have been developed for special industrial use.

Mains sub-miniatures on display should do much to reduce the size of electronic equipment both in the field of commerce and industry. The groups of battery subminiatures are specially suitable for portable lightweight electronic units. A range of all-glass photocells on the B8G and miniature B7G base offer great possibilities to all designers of industrial electronic control equipments.

Among an interesting group of thyratrons was a new miniature type 2D21 and a cold cathode type 1267. The latter tube should prove of great value in welding and industrial engineering timers, sequential process timers, alarm, fault and protective systems and remote control power switching.

Particular attention was given to an extensive range of electronic flash tubes suitable for commercial photography and numerous applications in industry and research. Tubes exhibited included the LSD3, see Fig. 26 which is widely used in lightweight portable flash equipment. There was the LSD2, microsecond flash tube for ultra-high speed photography; a new 1,000-joule tube specially designed for stage and film photography, and a special LSD8 tube for use in stroboscopic equipment.

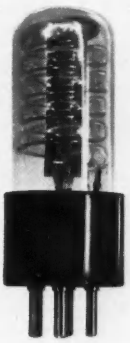

Fig. 26

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This new, light weight Industrial Truck is made entirely of 'Castle' fibre and will withstand rough usage. Standard size has 12 cu. ft. capacity & will carry 5 cwt. distributed load.

Supplied with one or three box section reinforcement bands.

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The first of the new series of hard glass V.H.F. valves, the TY2-125 triode and the QY3-125 tetrode, are now available. Both the triode and tetrode use thoriated-tungsten filaments and carbon anodes. The maximum frequency is 200 Mc/s.

James Neill & Co. (Sheffield) Ltd. have shown for the first time the " Eclipse " Magnetic Base for use with dial gauges. This unit ensures perfect rigidity of the tool during use, and the patented flux controlled base can be applied to either horizontal or vertical surfaces as well as upon cylindrical surfaces, such as milling machine arbors, or to the spindles of drilling machines. The design of the base allows it to be used upon its base or upon two of its sides, as well as upon a " Vee " section, the latter allowing its use upon cylindrical surfaces, and the magnetic force which holds the base in position is push-button controlled, permitting immediate release.

" Eclipse " Magnetic Chucks were once again featured in the full range of rectangular and circular models as well as other accessories.

For the users of Permanent Magnets and to the designer of equipment using magnets, the range of magnets on view was particularly revealing, for there were magnets for every branch of engineering shown in such a manner as to emphasise the wide development which has taken place during recent years in magnet design and magnetic efficiency by use of the wide range of alloys now available.

Supporting the display was a full range of " Eclipse " tools including Hack Saw Blades and Frames, High Speed Tool Bits, Piercing Saw Blades and Frames, Fret Saw Blades and Frames, Coping Saw Blades and Frames, Pin Vices, Pin Tongs, Pin Chucks, Pad Handles, Vee Blocks, Angle Plates, Surface Gauges and Scribers.

The Rheostatic Company Limited, Slough, Bucks., were exhibiting a very wide range of " SATCHWELL " temperature control devices including thermostats for water heater control, space temperature control, the control of primary firing devices such as coal stokers, oil burners, gas fired boilers and automatic gravity fed boilers together with their associated equipment. In addition, they were exhibiting motorised valves of the on/off type for the control of water and steam for central heating and industrial purposes. They were also showing a three-port modulating mixing valve which is designed to control the mixture of waters of two different temperatures to provide water at the third leg of the valve at any desired temperature. A further new device was a straight through type modulating valve which will have a particular application to the control of heater batteries used in air conditioning plants and to the control of steam to water heat exchangers.

Sigmund Pumps Ltd. exhibited at Birmingham some new additions to their range of pumps. This included the SIGMA Type " PR " Peripheral Pump. The need for small capacity high pressure pumps has, in the past, been largely met either by positive action or by multi-stage turbine type centrifugal pumps. This Company is now, however, able to offer for this type of duty the new range of " PR " Peripheral Pumps which will cover outputs up to 50 g.p.m., and with which one can obtain heads as high as 300-400 feet with one impeller.

Considered by many to be Sigmund's most important service to the industrial field is their range of SIGMA Process Pumps, which has been extended considerably in recent months, so that they can now offer a selection of pumps to the oil refinery and chemical industries which could in the past only be obtained from American sources. Concurrent with process pump production, Sigmund Pumps Ltd. have developed a complete series of mechanical seals. Rotating parts have been reduced to a minimum, thus departing from the conventional design, and gland leakage is completely eliminated whether the seal is required for high pressure or for vacuum work.

In recent months, this Company has made considerable additions to their range of split casing pumps, which now includes single stage single entry, single stage double entry, and two stage single entry types, covering outputs up to 6,000 g.p.m., and heads up at 560 feet.

Standard Telephones and Cables Ltd., exhibited their range of Low Power Radio Frequency Heating Equipments: Type IS, for preheating of moulding material, drying small quantities of materials, stiffening sheets of thermoplastic materials, etc.; Type 3S, for the setting or curing of synthetic glues, heating and welding of thermoplastics, drying and preheating of thermosetting plastics, etc.; Type A2402, Screened Electrode Assembly, suitable for use in conjunction with dielectric heating equipment when it is required to heat such materials as moulding preforms, sheet rubber, etc.; Type A2401, Dielectric Changeover Switch, developed to enable the r.f. output voltage to be switched to alternative electrode assemblies; Type 4M, for surface hardening small tools and gear wheels, for brazing and soft soldering details where speed and local heating are essential to ensure a clean and perfect joint, for annealing, and for small quantity melting; Type A2400, Induction Changeover Switch, applicable for example, in the brazing of cycle handlebars. Typical work coils for induction heating were also shown, the services of this Company's specialists being available to advise upon the design of coils to suit specific requirements.

J. P. Udal, Birmingham, were exhibiting their Guards for Presses, at Birmingham. Their " Fastrip " Guard for Power Presses was originally designed in view of customers' criticism with regard to the loss of output when using Hand Operated Interlock Guards, and it has since been realised that, apart from increased output with the " Fastrip " Guard as against the Hand Operated Interlock Guard, the " Fastrip " Guard has added safety features which in themselves are very important. The " Fastrip " Guard was shown on Hordern Mason & Edwards, Taylor & Challen, Bliss and Turner Brothers Presses, and is demonstrated under actual working conditions. Interlock Guards for Hydraulic Moulding Presses were also shown.

Wessex Industries (Poole) Ltd., were exhibiting their new " Wrigley " Lightweight Motor Trucks, which give every indication of filling a definite want for a cheaper and economical Motor Truck. These were shown, in the full range, together with Hand

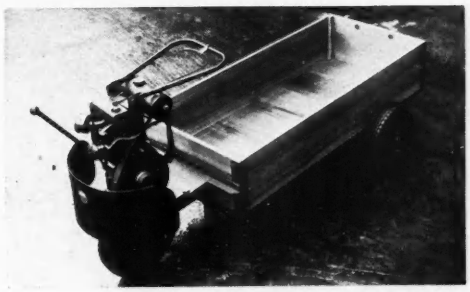

Fig. 27

Trucks and Trolleys, and Motor Hoes. " Low Load " Truck, Type 201 with a boarded platform 5 ft. 3 in. × 2 ft. 4 in. and a loading height of only 12 in. is shown in Fig. 27.

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On the 'temper' of a steel spring rests its capacity for service. Mass production of a design is easy, but to reproduce the same 'character' in each one calls for the greatest skill and experience. Next time you order springs, remember 'temper' is all-important. That will remind you of our name, and of our service.

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No longer need pressure gauges be brought back to workshops for testing. This compact and easy-to-carry calibrator will test them in a few minutes on site—with the highest possible accuracy. By placing weights up to a maximum of only 50 lbs on the left hand plunger, gauges can be tested to 400 lbs per sq. in: whilst the same weights placed on the right-hand plunger will test up to 4000 lbs per sq. in. A unique feature is that all wearing parts are replaceable by instantly interchangeable spares.

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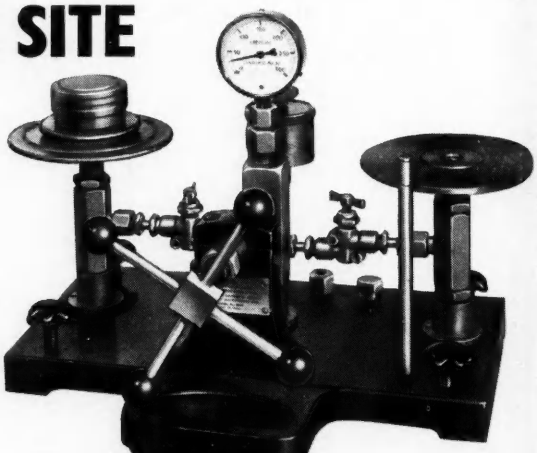

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NEWS OF THE MONTH

PERSONAL

Mr. A. S. Black, Manager of the Scottish branches of the General Electric Co., Ltd., has been appointed to the board of directors of the Company.

Mr. H. W. Bosworth, A.M.I.E.E., chairman and managing director of Lancashire Dynamo & Crypto Ltd., has been elected chairman of the council of the British Electrical and Allied Manufacturers' Association, 36-38, Kingsway, London, W.C.2.

Mr. C. V. Brewster has been appointed general manager and **Mr. D. L. Dennison,** sales manager of the Birmingham Tool & Gauge Co., Ltd., Soho Hill, Handsworth, Birmingham 19.

Mr. Mungo Campbell has been re-elected president of the North East Coast Institution of Engineers and Shipbuilders.

Mr. C. S. Couldwell has been appointed chief engineer of Consett Iron Co., Ltd., Consett, Co. Durham.

Mr. Arthur Dyson, M.I.C.E., has been appointed vice-chairman of W. G. Allen & Sons (Tipton) Ltd.

Sir Ernest Fisk has been re-elected President of the International Federation of the Phonographic Industry.

Mr. Kenneth Hall, managing director of the Northern Aluminium Co., Ltd., has been appointed President of the Council of the Aluminium Development Association.

Mr. H. V. Harris, secretary of Bennis Combustion Ltd., Little Hulton, Bolton, has been appointed to the board of the Company.

Mr. E. Johnson has been appointed managing director of the Atlas Diesel Co., Ltd., Beresford Avenue, Wembley, Middlesex.

Mr. R. C. Plowman, M.I.E.E., executive director of the British Thomson-Houston Export Co., Ltd., retired after 30 years' service with B.T.H. **Mr. E. V. Small,** previously director and general manager, has been appointed managing director and **Mr. J. N. MacDonald** has been appointed to the board as executive director of the British Thomson-Houston Export Co., Ltd. **Mr. H. E. Whetered** and **Mr. W. C. Scott** have been appointed managers Eastern sales (Middle and Far East, Africa, Australia and New Zealand) and Western sales (Europe, Russia and the Americas) respectively, with headquarters in London.

Mr. D. Pole-Welman, managing director of Foster, Yates & Thom Ltd., Blackburn, has been elected to serve on the North-Western Gas Board as a part-time member.

Mr. H. P. Potts, M.I.Mech.E., has been appointed sales director of B.S.A. Tools Ltd., Birmingham, Burton, Griffiths & Co., Ltd., Birmingham, and the Index Automatic Machine Co., Ltd., Redditch.

Mr. J. C. Proudfoot, Mr. W. H. Harding and **Mr. C. Mitchell** have been appointed works directors of Ambrose, Shardlow & Co., Ltd., Sheffield.

Mr. Jack Ridgeway, managing director of William Ridgeway & Sons Ltd., Sheffield, has been elected president of the Federation of British Hand Tool Manufacturers.

Dr. W. F. Riester, D.Sc. (Eng.), M.I.Mech.E., has been appointed to the board of directors as technical director of Sigmund Pumps Ltd., Team Valley, Gateshead 11. He is also in charge of Sigmunds' London office at 52, Grosvenor Gardens, S.W.1.

Mr. James Ritchie has been appointed shipyard manager of the Ailsa Shipbuilding Co., Ltd., Troon.

BUSINESS NOTES.

Air Control Installations Ltd., Ruislip, Middx., have opened a branch office at 19 Temple Street, Birmingham 2. Tel.: Midland 2546. **Mr. A. W. Evans, A.M.I.H.V.E.,** has been appointed manager.

Bennis Combustion Ltd. have formed a subsidiary company, Bennis Mechanizations Ltd., High Street West, Fenton, Stoke-on-Trent, to take over and develop the business of Richard Crittall Mechanizations Ltd. **Mr. F. C. Cooper** will continue to serve the new concern as director and general manager.

Blaw Knox Ltd. announce that their furnace equipment department is now operating from 7 Penley Court, Strand, London, W.C.2. Tel.: Temple Bar 3882. All other departments of the company continue at Clifton House, Euston Road, London, N.W.1.

The Woodall-Duckham Vertical Retort and Oven Construction Co. (1920) Ltd. has moved its head office to Woodall-Duckham House, 63-77 Brompton Road, London, S.W.3. Tel.: Kensington 6355.

INTERNATIONAL CONGRESS FOR HARBOUR ENGINEERING, ANTWERP, JUNE, 1949

On the occasion of the twentieth anniversary of the establishment of the Flemish Society of Engineers, Belgium (Vlaamse Ingenieursvereniging) an international congress for Harbour Engineering will be held in Antwerp from 16th to 19th of June, 1949.

MR. R. DE NAEYER, Secretary-General of the Belgian Ministry of Public Works will be President of the Congress, the activities of which will be grouped in four sections: (1) Harbour civil engineering; (2) Harbour machinery; (3) Handling and stocking perishables of either vegetable or animal origin; (4) Safety in harbours.

For further information apply to the Secretary General V.I.V., Torengebouw VIII, Schoenmarkt 31, Antwerp (Belgium).

NEW "TURBO-HEARTH" METHOD.

Basic principles for making steel rapidly in a new vessel called a "turbo-hearth" have been established by pooling the research of two major steel producers. The new method will permit the manufacture of steel of open hearth quality, low in phosphorus and nitrogen, by directing jets of air across the surface of white-hot liquid pig iron.

Announcement of this new American development was made today jointly by spokesmen of the two collaborating units, Dr. E. C. Bain, vice-president, for Carnegie-Illinois Steel Corporation, and H. W. Graham, vice-president, for Jones & Laughlin Steel Corporation, both of Pittsburgh. Credit also was given to C. E. Sims, of Battelle Memorial Institute, Columbus, Ohio, who assisted Carnegie-Illinois with research proposals. "The metallurgical feasibility of the process has been demonstrated," Dr. Bain and Mr. Graham asserted, "by the results of preliminary tests using small-scale units, these tests providing sufficient background for the design and construction of a full-sized furnace in which operating problems may be studied experimentally."

The turbo-hearth was developed in an attempt to combine the advantages of the basic open hearth and acid Bessemer processes. Turbo-hearth steel is made in a partly closed, side-blown vessel, lined with refractory similar to that used in the Thomas basic Bessemer process. The brick lining of the vessel is magnesia rather than silica. As in the Bessemer process generally, heat is supplied by chemical reaction of the blown air with carbon and other elements of the charge. Chemical action is fast and direct, but subject to controlled conditions which reduce the nitrogen and phosphorus contents, making turbo-hearth steel comparable to open hearth steel, hence suitable for more general use than the customary Bessemer steel.

The acid Bessemer and the basic Thomas processes have an advantage over other steelmaking processes, in that neither requires fuel from external sources. The choice between them depends

Cylinder copper-plated externally only, ready for carburising — an application of the Brailey Process.

Industrial applications of the Brailey Process include the deposition of chromium, nickel, cadmium, copper, silver, etc., for reclaiming worn or undersized parts or for protection against corrosion in service. Approved by all government departments. *The many different applications of the Brailey Process are covered in our 32 page booklet—available on request to all engineers.*

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Ermeto Shut-off and Flow Control Valves for oil-hydraulic pressures are now available. These valves are machined from steel bar-stock and are designed for hydraulic pressures up to 5,000 p.s.i. Reasonable quantities can be supplied either ex-stock or in 21 days.

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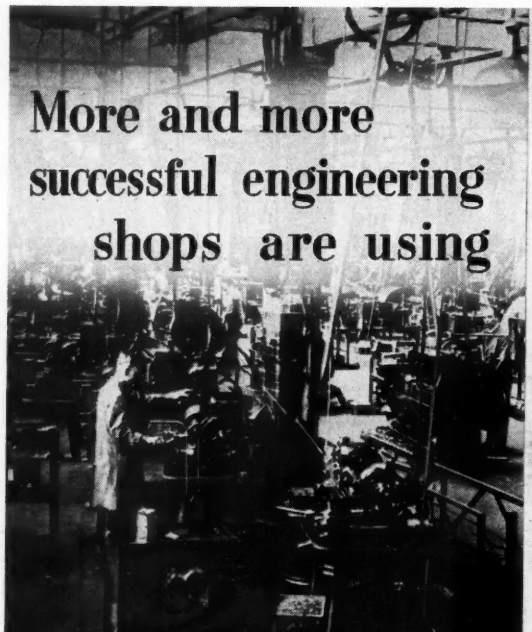

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upon the relative silicon and phosphorus contents of the iron that can be made from available ores. The reactions between the oxygen of the blast and the several elements in the liquid iron charge supply enough heat units to produce steel with sufficiently high temperature for subsequent finishing and teeming.

The high-silicon iron used in the Bessemer converter and the high-phosphorus iron in the Thomas operation are required largely because carbon, the predominating heat-producing element in the charge, is not completely utilized. Combustion of the carbon in the vessel proceeds only to the carbon monoxide stage, so that two-thirds of the potential heat from the combustion of carbon is lost when the monoxide burns to carbon dioxide in the air outside the vessel. Conservation of this heat has been practiced for years by producers of small, intricate castings. This has been accomplished with low-phosphorus iron and without excessive silicon by introducing the air at the side or on the surface of the converter charge in such a manner that sufficient oxygen is available to burn carbon to carbon dioxide inside the vessel.

These facts suggested that it should be possible, by directing jets of air on the surface of liquid iron to utilize the extra heat from the more complete combustion of carbon to produce steel from basic pig iron. The turbulence created by the impinging air could be expected to accelerate greatly the refining reactions.

Possible applications for such a process would be: (1) The production of low-nitrogen steel in an acid-lined vessel; (2) The utilization of conserved heat to melt larger amounts of scrap, and (3) The use of a basic lining for producing steel low in both phosphorus and nitrogen contents.

Operating difficulties such as are generally associated with the development of a new process have not yet been completely overcome, the sponsors said. Experience gained during this work, however, has shown to their satisfaction that the turbo-hearth process is fundamentally sound and that operating problems are not insurmountable.

BOOKS

Tensor Calculus Fundamentals and Their Analytical Representation. (Grundzüge der Tensorrechnung in analytischer Darstellung). By PROFESSOR ADALBERT DUSCHEK, Ph.D., and AUGUST HOCHRAINER, Dr. Sc. Techn. Part I: Tensor Algebra, 129 pp., 26 Figs. Publishers: Springer Verlag, Vienna, 2nd Edition, 1948. PRICE: 13/6.

This book has been written by a mathematician and an electrical engineer, and its purpose is to show that tensorial methods can be used in any types of problems involving co-ordinate geometry. Whereas vectors are symbols which "move freely in space, and only maintain their direction and orientation," tensors are intimately related to a system of co-ordinates and to the points in space, and lead to results which can immediately be translated into components and numbers. In this first volume, the authors introduce tensorial methods gradually, and apply them to various problems of plane and solid geometry. A certain knowledge of determinants and matrices is assumed. Each section is completed by a number of illustrative problems and examples. Volume II, which will be published shortly, will deal with tensor analysis and the theory of surfaces, while Volume III will give applications of tensor analysis in elasticity, hydrodynamics, electricity, and acoustics.

Mechanical Vibrations. Their Calculation, Design Effects and Suppression. By R. A. Collacott, Ph.D., B.Sc. (Eng.), A.M.I.Mar.E. Publishers: Sir Isaac Pitman & Sons, Ltd., Parker Street, Kingsway, London, W.C.2. 130 pp. 82 figures. PRICE: 15/-.

This is an excellent little book dealing with a most topical subject with praiseworthy clarity and simplicity. Whilst there are quite a few contemporary books on the subject, Collacott has devoted much more care and space to treating the practical aspect of mechanical vibrations, particularly their measurement, than other authors.

Rubber in Chemical Engineering. 2nd Edition. By H. P. STEVENS, M.A., Ph.D., F.R.I.C., and M. B. DONALD, M.Sc., A.R.C.Sc., M.I.Chem.E., F.R.I.C. 82 pp. Publishers: The British Rubber Development Board, Market Buildings, Mark Lane, London, E.C.3.

First issued in 1933, reprinted in 1939, the book has been very thoroughly re-written to set out the present position of rubber as applied to the chemical and allied industries. Opportunity has been taken to replace a number of the old pictures with more modern illustrations and the considerable technical advances which have taken place during the past decade have been incorporated in the text. Natural rubber is, of course, the material of choice, but, to ensure completeness, reference has been made to "synthetic" rubbers.

The first chapter deals with the properties of rubber—raw, soft and hard, with vulcanite or ebonite, tables giving their resistance to acids and alkalies, their mechanical and physical properties, etc. Chapter Two deals with linings of rubber sheeting, giving the present practice in methods of application, including testing and design. The all-important question of bonding is dealt with in detail and a section of this chapter is devoted to non-metals such as rubberised cements. Thermosetting and thermoplastic adhesives, including Factice and a number of alternative methods are also outlined. The chemical resistance of rubber is covered in Chapter Three and detailed tables give the resistance of hard and soft rubbers to an amazing number of chemicals—inorganic acids, inorganic salts and alkalies, organic materials, etc.

Compounding ingredients are also treated and reports of actual usage are given on hydrochloric, sulphuric and phosphoric acids, alkalies and gases. The chemical resistance of chlorinated rubber to specific chemicals is similarly dealt with in extensive manner and a useful bibliography of 100 references concludes this useful and informative Section. Chapter Four describes rubber paints,

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cements and varnishes, using oxidised, sulphonated and chlorinated rubbers. Rubber hydrochloride, and latex are also included. Specific uses of rubber in chemical engineering are described in Chapter Five—pipes, valves, plugs, chutes, conveyors, pumps, centrifugals, evaporators, etc., etc., are all covered in an informative manner. Rubber usage in electrochemistry and in elastomeric engineering are also outlined and a section deals with vibration and noise absorption. Well indexed, the book does much to show that rubber, already indispensable in the laboratory, is rapidly becoming a material of primary importance in chemical engineering. —*J.L.*

"Werkstattkniffe" Book Series. Publishers: Carl Hanser Verlag, Munich 27. No. 2. **"Schmierung,"** by DIPL. ING WALTER REUSCHLE. 4th Edition, 1948, 76 pp., 32 Figs. PRICE 3.40 DM.

This new and revised edition of the booklet on "Lubrication" contains some additional material as compared with previous editions and actually goes beyond the scope indicated by the title, inasmuch as it includes chapters dealing with coolants, cutting oils, oils for steel hardening, and with oil as a working medium, such as hydraulic oil. Another chapter deals with transformer oils.

The bulk of the book is of considerable interest not only to lubrication engineers generally, but also to the maintenance engineer in charge of any particular prime mover or process machinery. Good illustrations, number of tables giving specifications and treatment, and such aids to lubricant selection as nomograms, are to be found in the book.

No. 16. "Erprobte Werkzeuge," by LUDWIG MEISSLER. 1st Edition, 1948, 87 pp., 154 Figs. PRICE: 4.20 DM.

This is an extremely useful and well illustrated book dealing with tool design as applied to small metal pressings. Special tools of the type described in the book are concerned with the production of small and medium size parts from sheet of up to a thickness of 2 millimetres. Such tools are generally made by experienced toolmakers, mostly on the basis of hand sketches. It is the experience, the basis and skill of the tool-maker that count in the production of successful tools of this type. The present work is based on such experience and the selected examples have all proved successful in the case of thousands of components, such as those used in sewing machine attachments and the like. Cutting, bending, forming and riveting are dealt with in a manner likely to be useful for most people engaged on presswork.

No. 17. "Vorrichtungsbau," by H. E. SCHEIBE. 1st Edition, 1948, 84 pp. 76 Figs. PRICE 3.80 DM.

In this book on jigs and fixtures, the author, who is one of the principal engineers at the B.M.W. works, Munich, limits himself to useful examples without attempting to present a textbook on the subject. The examples selected have a bearing on economical mass production, that is, they indicate how the least chucking time can be obtained. Special emphasis has been laid on the necessity of employing standardized parts conforming to German DIN and AWF standards. In the reviewer's opinion, this book would amply repay study by our production experts, especially by those engaged in motor car production. This is even more the case because more is conveyed by the book's illustrations than by its text, thus, practically eliminating the language difficulty. The book is divided into the following chapters: Chucking, Lathe Jigs, Boring Jigs, Milling Jigs and Fixtures, Grinding Jigs, Dividing Jigs.

BRITISH STANDARDS

*Copies of British Standards may be obtained from the British Standards Institution, 24 Victoria Street, London, S.W.*1.)

Methods for Sheet Metal Pattern Development (B.S. 1549, Part I: 1949).

Notwithstanding the volume of sheet metal goods produced in this country the methods preferably to be employed in pattern drafting are insufficiently known.

The standard, of which Part I is now issued, deals primarily with assemblies in which, by reason of size, simplicity of outline or thinness of material, the neutral line may safely be ignored. In more complex structures where high dimensional accuracy is essential bend allowance, dependent upon the neutral line, is a factor of importance and is under experimental investigation. Part 2 of the standard will in due course deal with the development of such shapes.

The standard as now issued, deals with three methods of development for single curvature work by Radial line, Parallel line and Triangulation.

Twenty illustrative geometrical and calculation examples are included. PRICE 3/–.

Rolled Brass & Naval Brass Condenser Tube Plates (B.S. 1541: 1949).

The standard gives the chemical compositions for both types and covers dimensions and weight of plates. It also includes requirements for provision of test samples and details of making tensile and bend tests. PRICE 2/–.

Control Cocks for Low Pressure Gas (B.S. 1552: 1949).

This British Standard deals with control plug cocks of copper base alloy with nominal sizes ½ inch to 2 inches B.S. pipe thread used mainly for meters for low pressure gas. The standard is limited in scope to plug cocks with separable keys in occasional use.

Cocks made from castings and hot pressings are both recognised The dimensions of plug squares, taper of plug, length of keys and dimensions of sealing surfaces have been standardized. Tests on the completed cocks are specified.

Illustrations of three types of cocks and of 90° and 180° niting washers are included.

There is also an appendix on the effect of a reduced gas-way 2/3rds of the cross sectional area of the nominal bore of the tubing. In this appendix the maximum resistance to gas flow through a typical meter and control cock with reduced area plug is given. PRICE: 2/–.

Equipment for Eye and Face Protection During Welding (B.S. 1542: 1949).

This Standard is complementary to B.S. 679, Protective filters for welding. It specifies standard dimensions for filter glasses, together with the requirements for essential points of design and manufacture and the materials which may be used for this equipment. The Standard covers spectacles, goggles, facemasks, helmets, handshields and screens; it includes recommendations on the use of these types of equipment and also deals with the question of screening welding operators to avoid injury and distraction to other workers in the shop.

Pictures of typical equipment are included in this Standard. PRICE: 2/–.

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